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SOUND-PROOF PARTITIONS

AN INVESTIGATION OF THE ACOUSTIC PROPERTIES OF VARIOUS BUILDING MATERIALS, WITH PRACTICAL APPLICATIONS

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SOUND-PROOF PARTITIONS

I. Introduction

1. *Purpose of Investigation.*—The demand for quiet rooms in hospitals, hotels, and office buildings, the desirability of insulating music studios and other rooms where disturbing sounds are produced, and the necessity for solving other problems for the control of noise have led to repeated requests from architects and builders for reliable information on effective methods for insulating sound. Although present knowledge of the subject is incomplete, nevertheless, on account of the pressing need for guidance in such matters, it is thought desirable to collect and present the available information in a systematic way, giving the methods and results of various investigations relating to the action of materials on sound, describing practical installations of soundproofiing, and setting forth in accordance with existing knowledge recommendations that may be applied where sound insulation is wanted.

Before 1915 little was known definitely about the subject and cut-and-try methods were used when soundproofing was desired. These cases were usually isolated and few published accounts are available, so that little progress was made. Since 1915 the problem has engaged the attention of scientists who have conducted investigations in the light of the known phenomena of sound and obtained results that are now being applied in practical constructions.

2. *Acknowledgments.*—In addition to the published accounts of investigations, to which proper reference will be made, acknowledgment is due to the Armstrong Cork Company, the United States Gypsum Company, and the Associated Metal Lath Manufacturers for their financial support and coöperation in the conduct of special investigations.

II. ACTION OF SOUND WITH APPLICATIONS TO BUILDINGS

3. *Origin of Sound.*—Sound of a definite frequency consists of a regular series of alternate compressions and rarefactions generated by a vibrating body and progressing in spherical waves in the surrounding medium. Figure 1 pictures the waves in air sent out by the human voice through a megaphone.

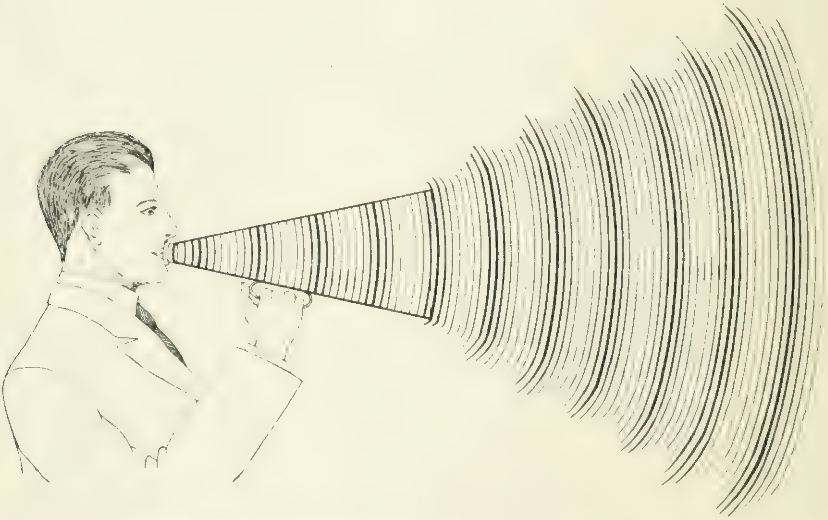

FIG. 1. DIAGRAMMATIC REPRESENTATION OF COMPRESSIONS AND RAREFACTIONS (SOUND WAVES) SENT OUT FROM A SOURCE OF SOUND

The vibrations in this case are due to the vocal cords which are set into rapid oscillations by a stream of air from the lungs. An unbalanced motor may cause its supporting base to shake, thus imparting to the surrounding air small motions that are propagated as sound waves. A case for consideration in buildings is the generation of sound waves when a wall or floor is set in vibration by a motor, an elevator, or other agency.

4. *Amplitude of Sound Vibrations.*—The amplitude of vibration in sound waves is small, according to estimates varying from 0.00000005 in. for a sound barely audible to 0.004 in. for a loud

sound.* A very small motion of a building partition† will therefore be sufficient to generate in air a sound that may be detected by the ear. Thus one of the difficult problems in sound insulation of buildings is to reduce the motions of walls as far as possible.

5. *Propagation of Sound.*—Sound waves set up by vibrating bodies are propagated through the surrounding medium—solid, liquid, or gaseous—with a considerable velocity, *v*, depending on the elasticity, *E*, and the density, *d*, of the medium, according to the formula: $v = \sqrt{E}\; d$. The values of the velocities for a few media are given in Table 1.‡

TABLE 1

VELOCITY OF SOUND IN VARIOUS MEDIA

Medium	Velocity of Sound at 0 Deg. C.
Air	1088 ft. per sec.
Water	4728 ft. per sec.
Pine Wood	10900 ft. per sec.
Brick	11980 ft. per sec.
Steel	16360 ft. per sec.

An inspection of the data given in this table shows that sound travels very fast, about one-fifth of a mile per second in air, and about three miles per second in steel. A sound traveling in the steel structure of a building 260 feet high would require only 260 ÷ 16360, or 0.0159 second to pass from the basement to the roof.

6. *Action of Materials on Sound.*—When sound waves in one medium encounter a second medium with a different elasticity or density, their regular progression is disturbed. Part of the energy is thrown back in the form of reflected waves, part is absorbed in the second medium, and part is transmitted—the relative amounts depending on the changes in elasticity and density of the second medium compared with the first, that is, in accordance with the change in the velocity of sound.¶ (See Fig. 2.)

---

* Shaw, P. E. "The Amplitude of Minimum Audible Impulsive Sounds." Proc. Roy. Soc., Vol. 76A, p. 360, 1905.

† Hall, E. E. "Graphical Analysis of Building Vibrations." Elec. Wld., Vol. 66, p. 1356, 1915.

‡ Smithsonian Tables, Tables 80 and 81.

¶ Rayleigh, Lord. "Theory of Sound." Vol. 2. Sections 270-272. Equations are given with discussions for different media.

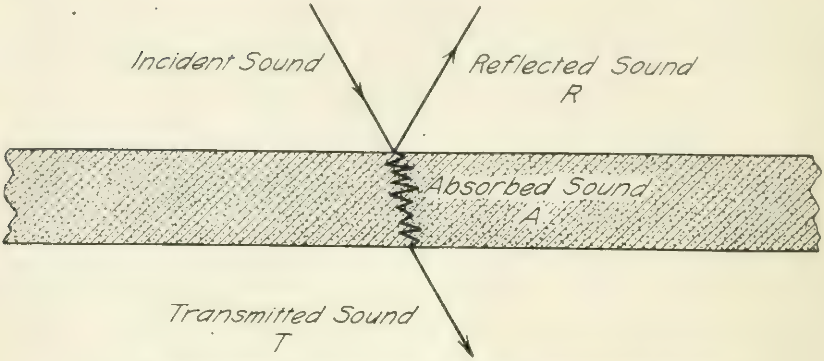

FIG. 2. REFLECTION, ABSORPTION, AND TRANSMISSION OF SOUND

A porous material like hairfelt presents little resistance to sound. The reflection is small, but the absorption in the porous channels may be quite large. What is not reflected and absorbed is transmitted. If the sound waves generated in a room meet solid plaster walls of sufficient rigidity they will suffer a reflection of over 99 per cent because of the large change in the elasticity and density between air and solids. If a ventilator opening is encountered instead of a wall there is no change in the medium and the waves progress with little hindrance through the continuous air passage, being confined in the ventilation duct by reflection from the metal walls. In a similar way sound vibrations generated in the solid matter of a building structure are confined to the structure by almost total reflection at the air boundaries and will pass with little interruption through the continuity of steel and concrete to distant parts of the building. These vibrations may be converted into sound waves in air where a wall or other structural member is set into sympathetic lateral vibration.

7. *Absorption of Sound.*—When air passages in which sound is passing become small in cross-section, friction that converts the wave energy into heat occurs between the sides of the passage and the oscillating air particles. Sound entering a small crack in a thick wall may be completely absorbed before emerging on the other side. The channels and interstices in carpets, hairfelt, and other porous materials, act in the same way in the absorption of sound energy.*

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* Rayleigh, Lord. Behavior of Porous Bodies in Relation to Sound. "Theory of Sound," Vol. 2, Section 351.

The absorption and transmission of sound vary with the thickness of the absorbing material but not in direct proportion. For example, if 1 in. of hairfelt stops 10 per cent of the incident sound, 2 in. will stop 19 per cent, 3 in. 27 per cent, etc.; that is, the intensity of transmitted sound decreases according to the recognized exponential law: $i = i_0 a^{-x}$, where i_0 and i are the respective intensities of the sound that enters and is transmitted by the material, a is a constant, and x the thickness of the material.

The absorption of sound is an essential factor in the solution of sound insulation. It is not sufficient to reflect and scatter sound waves, for the energy cannot be destroyed in this manner; it must be absorbed, that is, converted by friction into heat energy.

8. *Transmission of Sound.*—Sound waves in air may be transmitted through an obstructing medium in three ways. First, they may pass through the air spaces of a porous material. Second, they may be transmitted by modified waves in the new medium. In this process sound compressions and rarefactions progress rapidly through the air, moving the molecules successively as they pass in somewhat the same way as a gust of wind blows the separate stalks in a wheat field. On reaching a solid partition the forward motion is hindered, particularly if the molecules of the new material are massive and resist compression. In this case most of the energy is reflected and only a small proportion progresses through the wall. On meeting a further discontinuity of material, such as wood or air, the waves are again affected, until finally a part of the energy emerges. Third, sound may be transmitted by setting a partition as a whole in vibration. The partition then acts as an independent source of waves, setting up compressions and rarefactions on the further side and giving a sort of fictitious transmission. If the partition is rigid and massive the vibrations are small and very little sound is transmitted; if the partition is thin and flexible a considerable amount of energy is transferred.[*] Usually in building constructions the partitions are complex, as for example plaster on wood lath and studding. In this case the plaster areas between the studding act in a manner similar to drum heads and transmit sound. Hard

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[*] Rayleigh, Lord. "Theory of Sound." Loc. cit. See also:

Jäger, G. "Zur Theorie des Nachhalls." Sitzungber, der Kaisl. Akad. der Wissenschaften in Wien. Math. Natur. Klasse, Bd. CXX, Abt. 2a, 1911 plaster on wire lath presents a different surface with a modified action on the incident sound.

The transmission of sound involves a number of phenomena and is not a simple matter. It depends essentially on the character of the structure through which sound is transmitted and can be calculated only for simple cases of homogeneous materials of known constants.

III. INSULATION OF SOUND IN BUILDINGS

9. *Two Types of Sound in Buildings.*—Two types of sound should be considered in the problem of insulation in buildings. One type includes sounds that are generated in the air and that progress through the air to the boundaries of the room; the other is composed of compressions generated in the building structure by motors, elevators, and street traffic. The insulation of these disturbances is best accomplished by considering the actions of sound described in the preceding discussion. Several suggestions for soundproofing are given in the paragraphs immediately following.

10. *Insulation of Sounds in Air.*—Sounds of moderate intensity such as those generated by the human voice or a violin may be stopped with comparative ease if the walls of the room are continuous and fairly rigid. The more vigorous sounds of a cornet, trombone, etc. would require especially heavy walls or else double partitions. Any breaks in the walls for ventilators, pipes, or doors should be guarded by effective insulation.

11. *Insulation of Building Vibrations.*—Compressional waves generated in the building structure pass readily along the continuity of solid materials, and, as they have more paths for escape, are more difficult to insulate than sounds in air. Moreover, they may create trouble when they cause a wall or floor to vibrate. The insulation is based on the same procedure as that used for air sounds; namely, to interpose a new medium differing in elasticity and density. An air space in masonry would be effective if not bridged over by solid material, but since this is impossible for ordinary building constructions as the weight of bodies necessitates contact for support, an approximate insulation is sought by using air-filled substances like dry sand, ground cork, hairfelt, or flax, that possess but little rigidity but are capable of sustaining a floor or a partition that is not too heavy.

12. *Need for Experimental Data.*—The preceding discussion indicates from the standpoint of theory how sound waves will act under given conditions and how they may be controlled. Calculations may be made for a few simple cases of more or less homogeneous materials, but for the complex structures usually found in buildings

the present theory is inadequate to give trustworthy information. In view of this situation the procuring of data by direct experimental tests is quite desirable, particularly for the transmission of sound through materials both homogeneous and complex. Information about the absorption of sound is also needed. The following section gives in historical order descriptions of experimental investigations that have been completed or are now in progress.*

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* See condensed account of experiments performed previous to 1909 in the article: "Schalleitung der verschiedenen Körper," in Winkelmann's "Handbuch der Physik," Vol. 2, pp. 554-562.

IV. EXPERIMENTAL INVESTIGATIONS OF ACTION OF MATERIALS ON SOUND

13. *Conduction of Sound by Solids.*—Tyndall describes an interesting and instructive experiment on the conduction of sound. He writes,[*] "In a room underneath this, and separated by two floors, is a piano. Through the two floors passes a tin tube 2½ in. in diameter, and along the axis of this tube passes a rod of deal (wood), the end of which emerges from the floor in front of the lecture table. The rod is clasped by india-rubber bands which entirely close the tin tube. The lower end of the rod rests upon the sound board of the piano, its upper end being exposed before you. An artist is at this moment engaged at the instrument but you hear no sound. When, however, a violin is placed upon the end of the rod, the instrument becomes instantly musical, not, however, with the vibrations of its own strings, but with those of the piano. When the violin is removed, the sound ceases. . . . . What a curious transference of action is here presented to the mind! At the command of the musician's will, the fingers strike the keys; the hammers strike the strings, by which the rude mechanical shock is converted into tremors. The vibrations are communicated to the sound board of the piano. Upon that board rests the end of the deal rod, thinned off to a sharp edge to make it fit more easily between the wires. Through the edge, and afterwards along the rod, are poured with unfailing precision the entangled pulsations produced by the shocks of those ten agile fingers. To the sound board (of the violin) before you the rod faithfully delivers up the vibrations of which it is the vehicle. This second sound board transfers the motion to the air, carving it and chasing it into forms so transcendently complicated that confusion alone could be anticipated from the shock and jostle of the sonorous waves. But the marvelous human ear accepts every feature of the motion, and all the strife and struggle and confusion melt finally into music on the brain."

Hesehus[†] investigated the sound conductivity of various solids. The samples were shaped so as to be geometrically similar and placed successively with the lower end resting on a resonating support. A vibrating tuning fork was then placed against the upper end and the relative conduction of sound was made manifest by the intensity of the resonance tone heard. He found that the conductivity for cylindrical bodies was directly proportional to the cross-section and inversely proportional to the length—the same relation that holds for conduction of heat and electricity. He found also that wood conducts sound much better parallel to the grain of the wood than at right angles. This fact suggests a construction for hindering the progress of sound waves, namely, a series of wooden slabs cut alternately parallel and perpendicular to the grain of the wood. If the slabs were separated by layers of felt and not nailed, the obstruction would appear quite efficient.

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[*] Tyndall, J. "Sound." Pp. 78-79.

[†] Hesehus, N. "Ueber das Schalleitungsvermögen der Körper." Jour. Russian Phys. Chem. Soc., Vol. 17, pp. 326-30, 1893. See also abstract in Fortsch der Physik, Vol. 43, p. 542, 1893.

14. *Experiments by Tufts.*—The transmission of sound by porous materials was tested in 1901 by F. L. Tufts.* For the initial experiment the porous material consisted of lead shot piled in a tube. This was exposed to the action of sound through a metal gauze that supported the shot. A thin rubber membrane closed the bottom of the tube below the gauze and by the amplitude of its vibration gave a measure of the sound waves transmitted. Cotton batting, felting, and other porous materials were tested in the same manner. Tufts concluded that the resistance of a porous material to sound was in the same proportion as the resistance to air currents.

In a later experiment he measured the transmission of sound by materials impervious to air.† Two thin discs of such material closed the ends of a bifurcated tube. Sound pulses, caused by a metal ball dropped on a pine board, impinged on the discs so that the transmitted waves passed on through the tubes to a junction point, where an observer could compare the loudness of the two sounds by alternately closing one tube and then the other. The materials were tested in pairs, each sample being in the form of a disc, 10.5 cm. in diameter. Table 2 gives the results obtained.

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* Tufts, F. L. "The Transmission of Sound Through Porous Materials." Am. Jour. of Sc., Vol. 11, p. 357, 1901

† Tufts, F. L. "Transmission of Sound Through Solid Walls." Am. Jour. of Sc., Vol. 13, pp. 449-454, 1902.

TABLE 2

TRANSMISSION OF SOUND BY MATERIALS IMPERVIOUS TO AIR

	Material	Thickness in cm.	Rigidity*	Transmitted Sound
1.	Lead	0.012	0.000106	Louder
	Glass	0.012	0.000053	
2.	Pine	0.65	0.000013	
	Leather	0.65	0.000212	Louder
3.	Reënforced brass	0.015	0.00008	Equal loudness
	Two thicknesses of cardboard	0.44	0.00008	
4.	10 sheets cardboard	0.70	0.0002	Equal loudness
	1 sheet cardboard	0.22	0.0002	
5.	Reënforced brass	0.015	0.00008	
	Brass with mass at center	0.015	0.0022	Much louder
6.	Cardboard	0.22	0.0004	Louder
	Cardboard with 34 gm. mass at center	0.22	0.0004	
7.	3 cardboards separated by air space	cardboards all cut from same sample		Louder
	3 cardboards in contact			

*The numbers in the rigidity column represent the displacement of the discs for a pressure of 1 gm. per sq. cm., the smaller values thus indicating the greater rigidities.

A study of this table leads to the following conclusions:

1. The more rigid glass is the better insulator;
2. The more rigid pine is the better insulator;
3. For the same rigidity the cardboard, 30 times thicker than the brass, transmits the same amount of sound;
4. For the same rigidity 10 sheets of cardboard transmit the same sound as one sheet;
5. The more massive disc with smaller rigidity transmits more sound;
6. For equal rigidity the more massive disc is the better insulator;
7. The cardboards in contact gave a greater rigidity and cut off more sound. The air spaces are thus not as effective as was supposed.

Inspection of these results, therefore, shows that rigidity is the deciding factor, the amount of sound transmitted as an elastic wave being negligible compared with the to-and-fro vibration of the material. For instance in case 4, where the two discs possess the same rigidity, one would expect the transmission to be greater through the single thickness of cardboard than through the built up disc with many reflecting surfaces; but the transmitted sound is the same for the two constructions.

A few additional experiments were conducted using an organ pipe for the source of sound, thus giving a sustained series of alternate compressions and rarefactions instead of a single pulse. It was found that the results might be influenced by the pitch of the note used, so that, if a disc happened to be set into resonant vibration, it would transmit sound to a greater extent than without such resonance.

15. *Investigation by Weisbach.*—A tuning fork giving 236 vibrations per second was used as the source of sound and placed on one side of a wall. A telephone connected to a galvanometer was placed on the other side of the wall and served as a receiver for the sounds transmitted through materials installed over a small hole in the wall.* Weisbach found that metal, wood, etc. vibrated, whereas cloth and porous materials did not. When the disc was free from resonant vibrations the transmitted sound was proportional to the mass of the disc per unit area. When resonance occurred, the energy transmitted by the lateral vibrations was so great that it obscured the other effects, the results depending on the size, structure, and method of fastening the material. He concluded that sound is transmitted through the pores of materials, also by longitudinal vibrations (elastic waves) and by transverse vibrations of the materials.

16. *Tests by McGinnis and Harkins.*—In this investigation, the source of sound was an organ pipe of 768 vibrations per second, the measuring instrument being a telephone receiver in series with a crystal rectifier and a galvanometer.† The materials tested were clamped over an aperture $\frac{1}{2}$ in. in diameter, through which the sound was transmitted. For porous materials such as calico and felt the investigators concluded that the amount of sound transmitted depended on the size and nature of the pores of the material, that is, on the true absorption. The non-porous materials like paper, oilcloth, and tin-foil transmitted less than one per cent of the incident sound when the lateral vibrations were eliminated. All the materials tested were quite thin and of small area.

17. *Work of Wallace C. Sabine.*—An investigation,†† fundamental and far-reaching in its scope, was inaugurated and partly

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* Weisbach, F. "Versuche über Schalldurchlässigkeit, Schallreflexion und Schallabsorption." Annalen der Physik, Vol. 83, p. 763, 1910.

† McGinnis, C. S. and Harkins, M. R. "Transmission of Sound Through Perous and Non Porous Materials." Phys. Rev., Vol. 33, pp. 128-136, 1911.

†† Sabine, W. C. "The Insulation of Sound." Tne Brickbuilder, Vol. 24, pp. 31-36, 1915.

carried out by the late Professor W. C. Sabine of Harvard University. A sound of measured intensity was generated in a room so that transmission took place through a partition under test. The sound was then reduced in intensity until an observer outside the partition could barely hear it. The test was repeated for other samples and comparative values deduced. The test rooms were especially adapted to confine the transmission of sound to the test material.

Table 3 gives results obtained in the preliminary investigation. Sheet iron and hairfelt were tested separately and in combination. The sheet iron because of its great rigidity and mass reflected much sound and transmitted but little. The reduction of transmission by the hairfelt was due largely to its absorbing power. The combination of the two in alternate layers gave the more effective insulation, the hairfelt serving to reduce further the small amount of sound transmitted by the iron.

TABLE 3

INTENSITY OF SOUND TRANSMITTED BY SHEET IRON AND HAIRFELT

Layers	Hairfelt	Sheet Iron	Sheet Iron and Hairfelt Combined
0	1 000 000	1 000 000	1 000 000
1	270 000	22 700	23 000
2	128 000	8 700	3 300
3	65 000	4 880	700
4	33 000	3 150	220
5	21 500	2 060	150
6	11 400	1 520	88

18. *Investigations by P. E. Sabine.*—Later experiments on sound transmission through windows and doors have been conducted by P. E. Sabine,* using methods similar to those of W. C. Sabine. He found that flexural vibrations play an important part in transmission. A plate of glass 3/16 in. thick transmitted less sound than a 3/16-in. composite structure of two 1/16-in. plates sealed together with a sheet of celluloid. Dead air spaces between double windows were not as effective as commonly supposed. The air space should not be bridged over by solid materials even at the edges of the glass. Inserting sound absorber between the glass plates increased the insulation. A steel door ¼ in. thick proved more effective than a solid oak door 1¾ in. thick or a refrigerator door 4 in. thick filled with heat-insulation material.

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* Sabine, Paul E. "Architectural Acoustics." Am. Arch., Vol. CXVIII, pp. 102-108, 1920.

Additional experiments were conducted to test the transmission of sound through flexible materials.[*] Results were obtained to determine to what extent the damping of sound by such materials as hairfelt, sea-weed, and asbestos is an effective compensation for lack of mass and stiffness. Numerous curves are given showing the transmission of sound for a range of pitches. Further investigation was planned to test more rigid materials, including plaster walls.

19. *Sound Deadening in Hospitals.*—Richard E. Schmidt gives a discussion[†] of sound transmission in buildings, paying particular attention to the character of building materials with acoustic properties suitable for use in hospital construction. The article gives sketches of rooms and photographs of buildings in Chicago that have sound-proofing features.

20. *Sound-Proofing Tests Made in Chicago Music Building.*—Table 4 pictures the construction and relative sound-proof efficiency of various types of building constructions. This table appears in the 1911 catalogue of the Corrugated Bar Company. It has been impossible to locate any description of the tests other than the information given in the table, but the results set forth are so thoroughly in accord with the conclusions reached in other investigations that the data are included as additional information.

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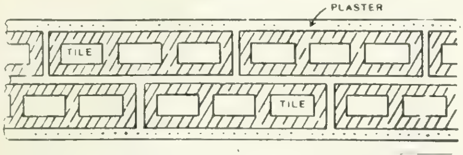
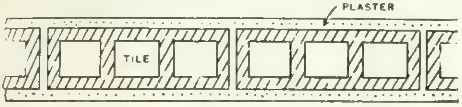
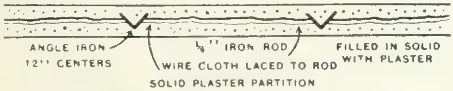
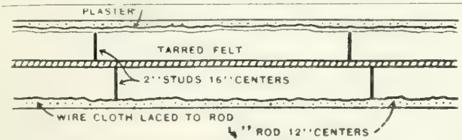
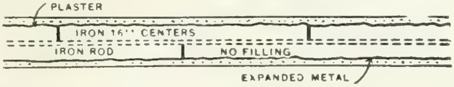
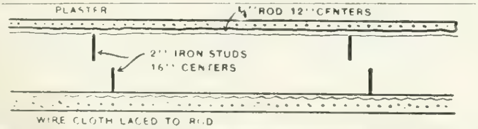
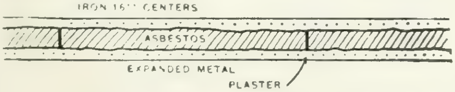
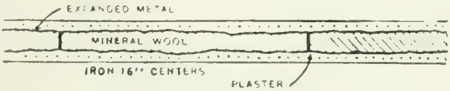
[*] Sabine, P. E. "Architectural Acoustics. The Transmission of Sound Through Flexible Materials." Am. Arch., Vol. 120, pp. 215, 266. 1921.

[†] "Sound Deadening in Hospitals." Surgery, Gynecology, and Obstetrics, Vol. XXXI, pp. 105-110, July, 1920.

TABLE 4

COMPARATIVE SOUND-PROOF TESTS MADE AT MUSIC BUILDING, CHICAGO, IN 1895.

TYPE OF PARTITION	Rank in Sound Proof Efficiency	
	1.	COMMON MORTAR (CEMENT)
	2.	COMMON MORTAR (CEMENT)
 ANGLE IRON 12'' CENTERS ¼'' IRON ROD WIRE CLOTH LACED TO ROD FILLED IN SOLID WITH PLASTER SOLID PLASTER PARTITION	3.	COMMON MORTAR (CEMENT)
 PLASTER TARRED FELT 2'' STUDS 16'' CENTERS WIRE CLOTH LACED TO ROD ¼'' ROD 12'' CENTERS	4.	ROCK PLASTER COMMON MORTAR (CEMENT)
 PLASTER IRON 16'' CENTERS IRON ROD NO FILLING EXPANDED METAL	5.*	ACME PLASTER COMMON MORTAR (CEMENT)
 PLASTER ¼'' ROD 12'' CENTERS 2'' IRON STUDS 16'' CENTERS WIRE CLOTH LACED TO ROD	5.*	COMMON MORTAR (CEMENT)
	5.*	ACME PLASTER COMMON MORTAR (CEMENT)
 EXPANDED METAL MINERAL WOOL IRON 16'' CENTERS PLASTER	5.*	ACME PLASTER COMMON MORTAR (CEMENT)

* Sound carried through due to metal connections.

V. APPARATUS AND METHODS USED IN PRESENT INVESTIGATION

21. *General Methods and Apparatus.*—The experiments on which this bulletin is based have extended over a period of seven years, but have been confined largely to three periods. The method used remained essentially the same throughout, but the apparatus and conditions were improved greatly for the later tests. Materials tested had considerable area—at least 3 ft. by 5 ft.—and varied in structure

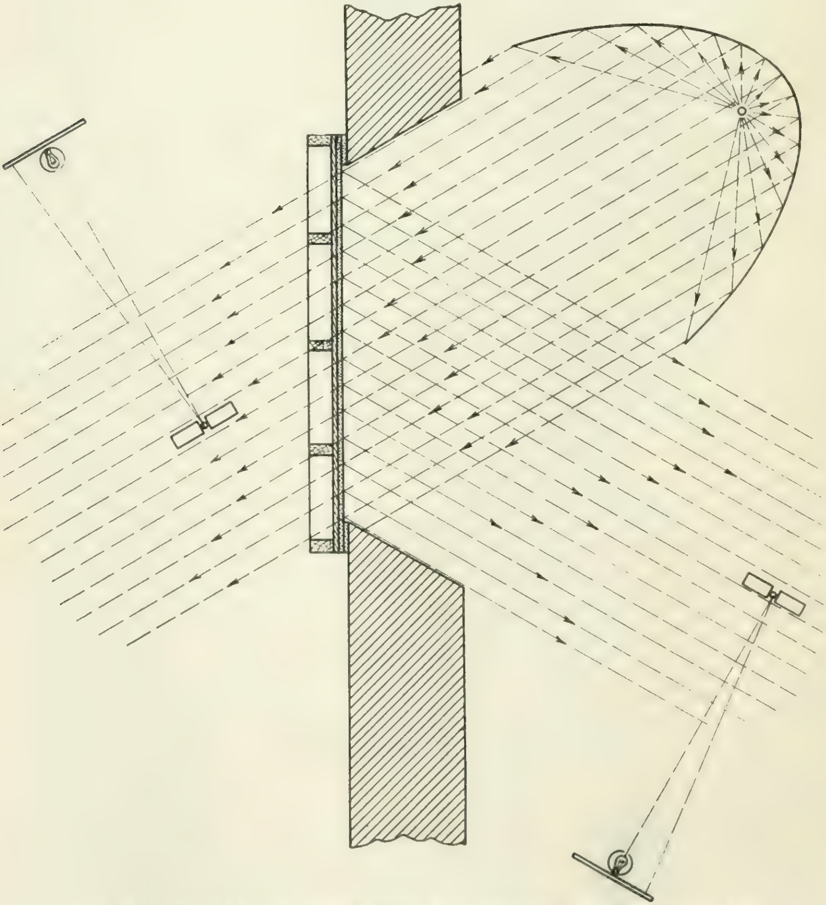


FIG. 3. DIAGRAM OF APPARATUS FOR TESTING TRANSMISSION AND REFLECTION OF SOUND

from thin flax and hairfelt to solid plaster partitions. An adjustable metal organ pipe blown by air from a constant pressure tank served as the source of sound. This was mounted at the focus of a parabolic reflector 5 ft. in diameter (see Fig. 3), so that a large percentage of the generated sound was directed in a parallel bundle against the partition under test. The sound was partly reflected and partly transmitted, the amounts of transmission and reflection being measured simultaneously by Rayleigh disc resonators placed in the paths of the sound.

22. *The Rayleigh Disc Resonator.*—The simplest form of this instrument consists of a tube closed at one end and open at the other. (See Fig. 4.) Sound waves of suitable frequency, coming to the instrument, set up standing waves in the tube with a considerable back and forth surging of the air at a "loop." A thin circular mica (or glass) disc suspended at this loop at an angle to the axis of the tube tends to turn flatwise to the alternating air currents. If a fine quartz thread is used to suspend the disc and the latter is placed at an angle of 45 deg. to the axis, a delicate means is provided for measuring sound, the amount of rotation of the disc varying from 0 deg. to 45 deg. depending on the intensity of the incident sound. This turning action is in accordance with the general principle that a flat object in a current tends to set itself flatwise to the current. Any turning of the disc is made visible by the movement of a beam of light reflected from its surface to a scale. Theory shows that

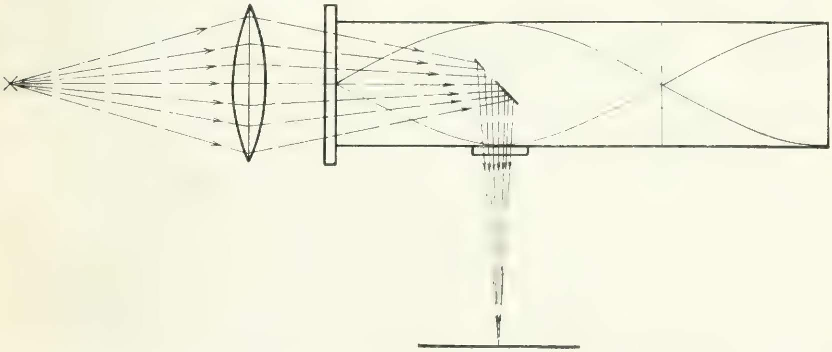


FIG. 4. DIAGRAM OF SIMPLE RAYLEIGH RESONATOR

the comparative intensities of two sounds acting separately on the instrument are given by the equation:*

$$\frac{I_1}{I_2} = \frac{\phi_1 \ sin \ 2 \ (\theta - \phi_2)}{\phi_2 \ sin \ 2 \ (\theta - \phi_1)}$$

where I_1 and I_2 are the intensities of the sounds, ϕ_1 and ϕ_2 the corresponding deflections of the disc, and θ (usually 45 deg.) is the angle between the normal to the disc and the axis of the tube.

Since the intensities of the transmitted sounds varied over a wide range during the tests, several resonators of different sensitivity were required to obtain the measurements. For the faint sounds transmitted by solid plaster partitions, it was necessary to develop a sensitive double resonator† of the proper frequency to obtain measurements. (See Fig. 5.) This latter instrument measured sounds of "threshold audibility," that is, sounds so faint that they could barely be detected by the ear. The resonators are much more sensitive than the ear in measuring sounds of different intensities. This is evidenced by the need of several resonators of different sensitivity to measure the same range detected by the ear. The hearing sensation is proportional to the logarithm of the intensity and is therefore quite insensitive.

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* Watson, F. R. "An Investigation of the Transmission, Reflection, and Absorption of Sound by Different Materials." Phys. Rev., Vol. 7, pp. 125-132, 1916.

† Rayleigh, Lord "Note on the Theory of the Double Resonator." Phil. Mag., Vol. 36, pp. 231-234, 1918.

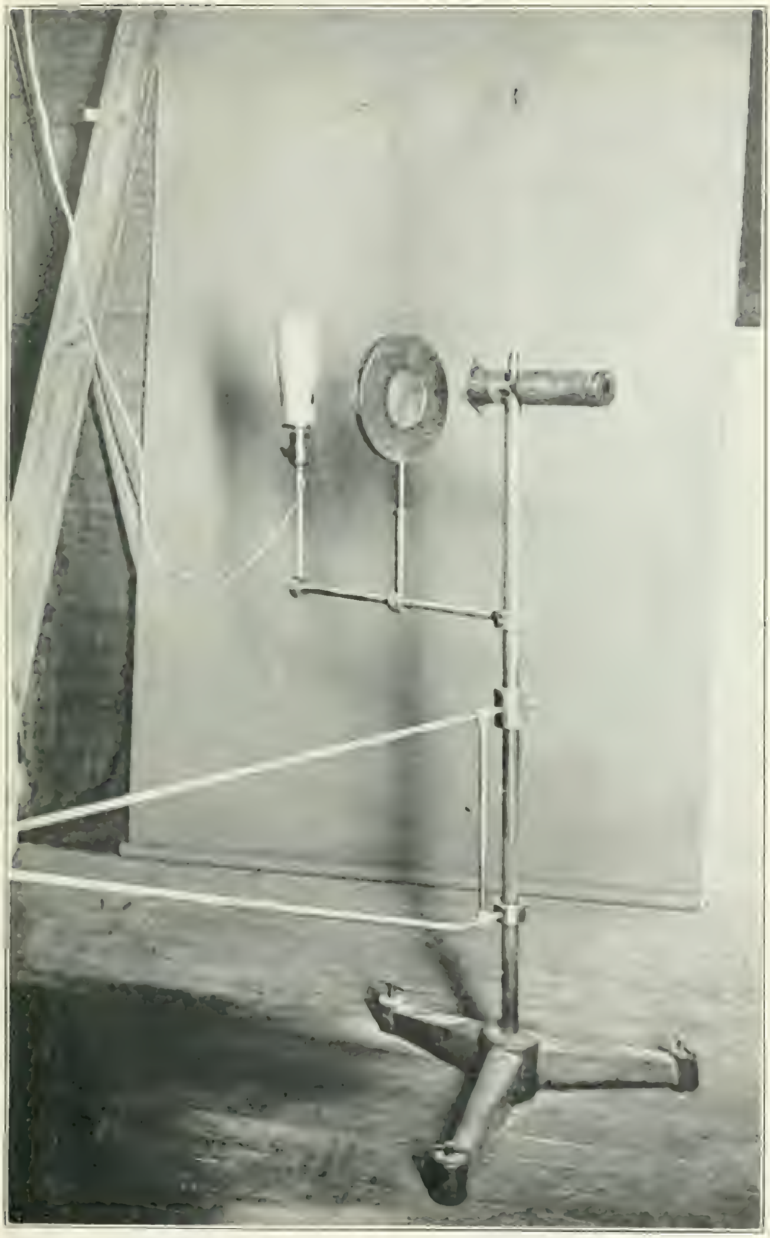

FIG. 5. SENSITIVE RAYLEIGH RESONATOR FOR MEASURING FAINT SOUNDS

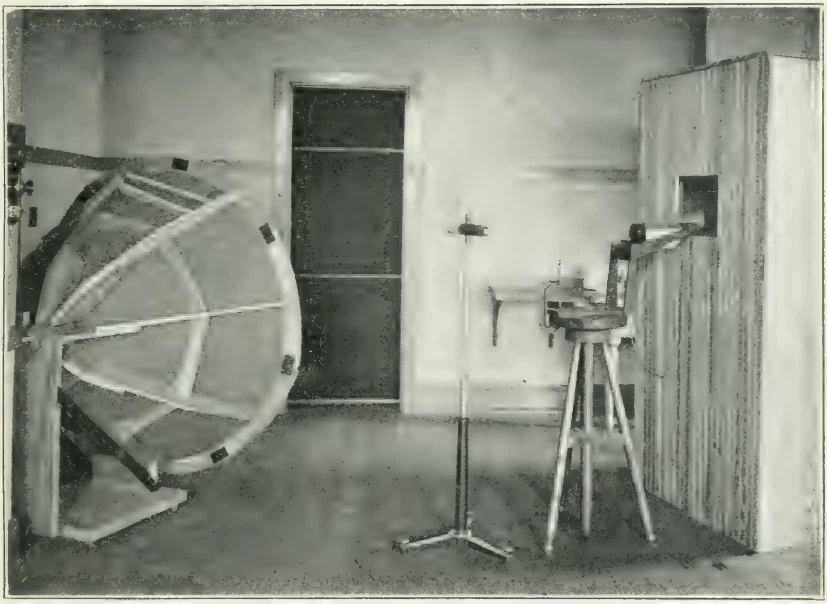

FIG. 6. APPARATUS USED IN INVESTIGATION

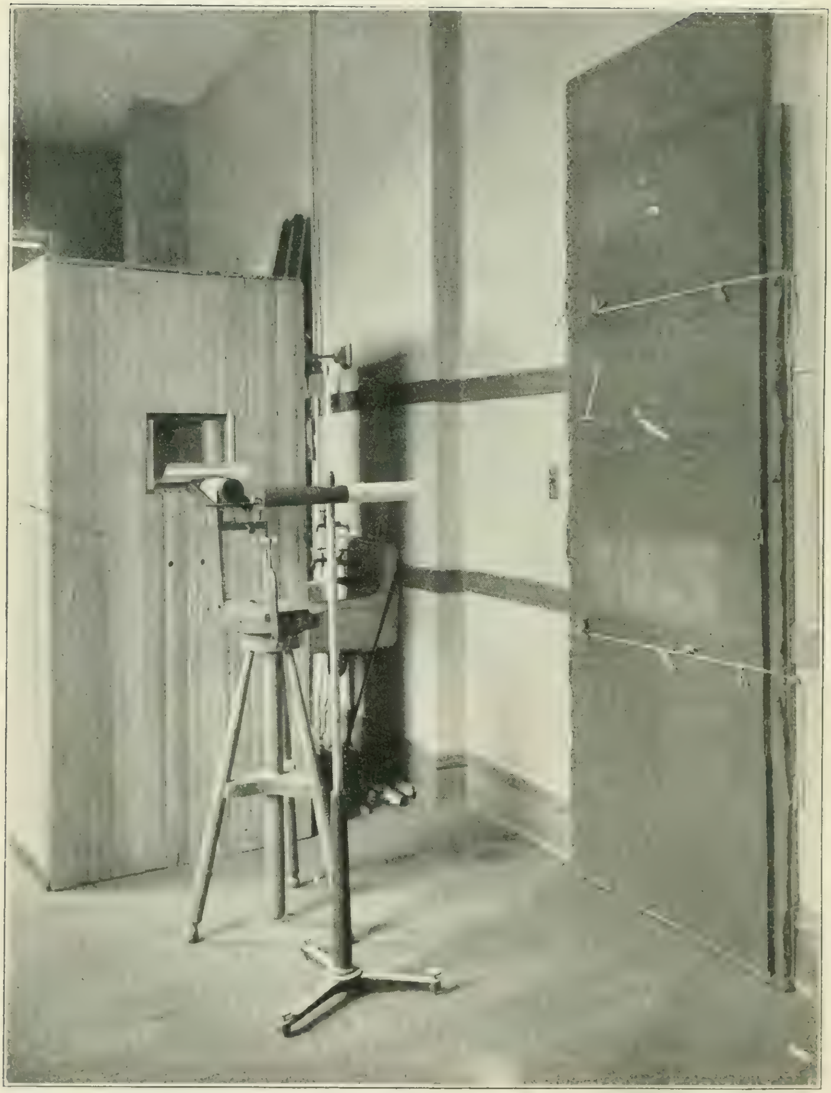

FIG. 7. APPARATUS FOR MEASURING TRANSMITTED SOUND

VI. PRELIMINARY INVESTIGATION

23. *Arrangement of Apparatus.*—Figures 6 and 7 illustrate the conditions for the first test. An organ pipe blown steadily by constant air pressure generated a sound that was transmitted through a doorway into an adjacent room where a Rayleigh resonator measured the energy. Measurements were taken, first through the open doorway, then with one panel of the material over the doorway, next with two panels, and finally with three panels; the deflection of the resonator being noted for each case. Since the tests were comparative, all conditions were maintained as constant as possible. Every article in the room was left undisturbed so that the interference pattern due to reflection of sound from various surfaces would remain unchanged and not affect the readings of the resonator. To eliminate the effect of the observer, who necessarily must have some freedom of motion, a small booth was built with a glass window. The observer could shut himself in this compartment and take readings through the window. The samples to be tested were fastened on similar frames of 1-in. cypress and mounted over the doorway by two ropes. A strip of hairfelt was installed around the door frame to prevent leakage of sound at the edges.

24. *Experimental Results Obtained.*—Table 5 gives results for the reflection and transmission of sound.

TABLE 5

TRANSMISSION AND REFLECTION OF SOUND

Material	Deflection of Resonator in cm							
	Reflection				Transmission			
Thickness in layers	0	1	2	3	0	1	2	3
Open Doorway	3.9				39.4			
½-in. Hairfelt		4.9	6.6	10.5		22.6	15.4	10.4
¼-in. Cork Board		15.7	22.0	22.6		7.9	3.75	2.9
¾-in. Cork Board		25.9	21.2	22.1		1.15	2.05	0.85
¼-in. Paper-lined Hairfelt		20.7	5.9	10.0		5.0	21.7	3.8
¾-in. Paper-lined Hairfelt		10.4	6.6	9.3		6.5	1.95	0.4
¾-in. Flax Board		22.5	20.0	20.0		2.25	0.55	0.1
¼-in. Pressed Fiber		23.2				0.32		
¾-in. Pressed Fiber						0.2		

25. *Discussion of Results.*—The results show the effects of different materials on sound waves. Porous hairfelt transmits con-

siderable sound, and the reflection is small. Other samples, impervious to air, reflect more and transmit less. A better conception of the results is obtained by considering the relative values of the transmission and reflection together with the absorption. Curves for these values may be plotted on the same scale by assuming that the open doorway transmits 100 per cent of the incident sound and reflects 0 per cent; also that the maximum reflection value (25.9 for the ¾-in. cork board) may be taken as 100 per cent. These assumptions are not rigorously correct, but they allow comparisons that are near the truth.*

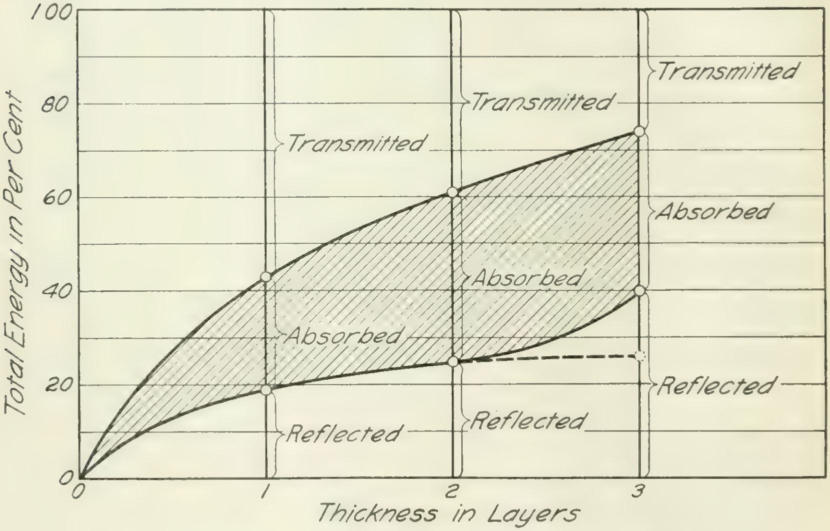

Fig. 8. Relative Amounts of Sound Reflected, Absorbed, and Transmitted by Hairfelt

26. *Action of Porous Materials.*—Fig. 8 shows the results for ½-in. hairfelt. Due to absorption of energy in the pores of the material the transmission decreases with increasing thickness, according to the law stated in paragraph 7. The reflection increases with increasing thickness but tends toward a constant value, indicating that the reflection does not take place entirely at the surface of a porous material but requires a certain thickness to give the maximum value. The absorption also appears in this diagram because the

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* Watson, F. R. Loc. cit.

incident sound ($I = 100$ per cent) is equal to the sum of the percentages reflected, absorbed, and transmitted.

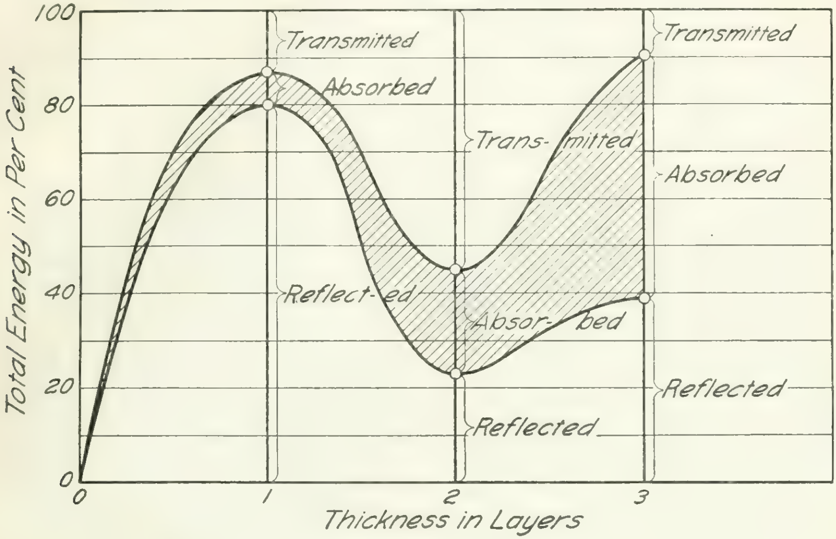

FIG. 9. TRANSMISSION, REFLECTION, AND ABSORPTION OF SOUND BY VIBRATING PARTITION

27. *Action of Materials Impervious to Air.*—Fig. 9 shows results that at first were quite puzzling. Two layers of paper-lined felt transmitted *more* sound than one layer; also, the reflection was *less* for two layers than for one. The explanation is easily seen when the transmission and reflection are plotted together. The two layers *vibrated* under the action of the sound so that the transmission was disproportionately large. An extreme case of this kind would exist if a material vibrated exactly as the air would if the material were not present. There would then be no resistance to the sound waves, hence no reflection, and all the sound would be transmitted.

28. *Conclusions.*—The information given in Figs. 8 and 9 is quite vital in the consideration of sound-proof structures. Fig. 8 shows the normal reflection, absorption, and transmission to be expected for compressional waves passing from one medium through another. Fig. 9 shows the anomalous effects set up when a layer of material vibrates as a whole. This is the usual case in building constructions where partitions vibrate more or less under the action of sound and give unexpected results. The obvious procedure to avoid this complication is to use rigid partitions and minimize the vibration.

VII. Investigation of Thin Plaster Partitions

29. *Arrangement of Apparatus.*—This set of tests was conducted in two basement rooms separated by a double wall consisting of two 9-in. brick partitions with separating air space. The walls enclosing the rooms were of heavy brick construction, the ceilings consisted of 4-in. concrete slabs, and the wood floors rested on concrete. A window cut through the double wall between the rooms was lined with cement plaster and allowed the sound generated in one room to pass directly to the partition under test. (See Figs. 3 and 10.) The reflected sound and the sound transmitted into the neighboring room were measured by Rayleigh resonators. The source of sound was an adjustable Edelmann pipe giving about 630 vibrations per second. The double doors connecting the rooms were padded with felt and arranged to shut as tightly as possible. The arrangements proved to be quite satisfactory for the experiments. Air currents were eliminated and disturbing outside sounds were reduced to a minimum. Any transmission of sound from one room to the other was thus confined to the sample under test.

30. *Materials and Method of Measurement.*—The materials tested varied in structure from porous hairfelt to plaster coating on wood lath. Since the measurements were to be comparative all conditions were maintained as constant as possible, the only changing factors being the samples under test. When taking measurements, the observer entered a small booth built for the purpose and closed the door. Each panel had the same size, 3 ft. by 5 ft., and was clamped in place over the window in the same manner by a wooden frame with iron bars and tightening nuts. Collapsible samples were held in place by wire netting. The range of intensities of sound transmitted was so great that two Rayleigh resonators of different sensitivity were used, intermediate values being taken by both resonators to give continuity of results. The reflected sound was measured simultaneously with the transmitted sound, thus giving check measurements, since any variation in conditions affected both the transmission and the reflection.

31. *Results Obtained.*—The plaster samples were constructed by an expert workman and allowed to dry before testing. They were

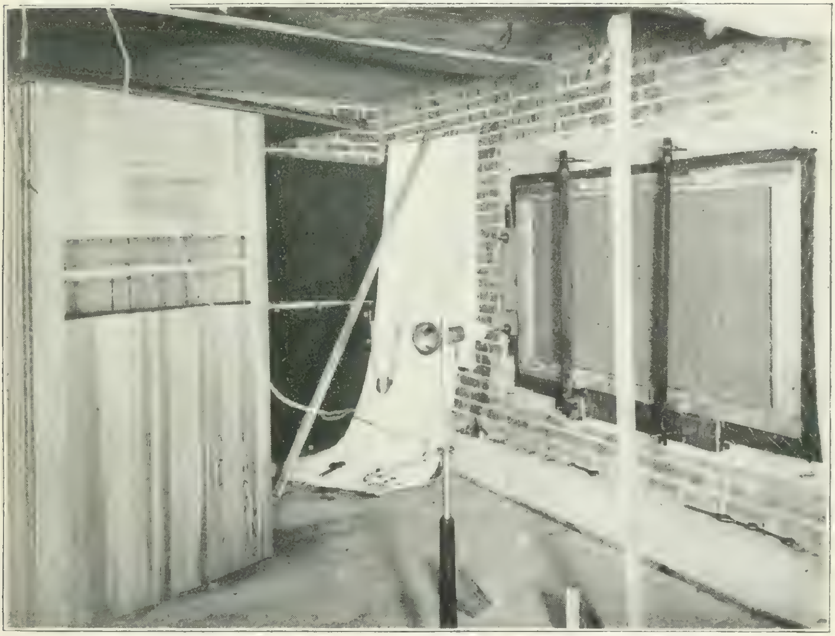

FIG. 10. VIEW OF TEST ROOM, SHOWING SAMPLE PARTITION CLAMPED OVER WINDOW

then tested one after the other under conditions kept as uniform as possible. The measurements were repeated five times, thus giving six observations for each result noted. In some cases of doubt, further measurements were taken. Table 6 indicates the results obtained. The first column of figures gives the actual deflections observed for transmitted sound, the three Sackett boards being tested by both resonators to establish the connecting ratio between the two sets of readings. The second column gives relative transmission values for the entire set of materials. These values were obtained by reducing proportionately the numbers in the second set so that the readings for the Sackett boards were the same in both sets. The third column gives deflections for the reflected sound, the same resonator being used throughout for these measurements.

32. *Discussion of Results.*—The results show that the porous, burlap-lined flax and the hairfelt transmit considerable sound and reflect little. Paper-lined materials—the Keystone Hair Insulator and Cabot Quilt—transmit less and reflect more. The Sackett boards are the most efficient sound insulators of the thinner samples, while in the plaster panels those containing gypsum plaster appear to be more effective in stopping sound. This is probably due to the fact that gypsum produces a stiffer, more rigid structure. Some estimation of the absorption of sound may be formed by remembering that the sum of the transmitted, reflected, and absorbed sound equals the constant incident sound. If the transmission and reflection are both small, as with the mineral wool for example, the absorption is correspondingly large. Thus with the plaster panels the reflection is extremely large, and the transmission and absorption correspondingly small.

Table 6

Deflections of Resonator for Transmitted and Reflected Sound

Material	Transmission		Reflection
½-in. Flax, burlap-lined	69.4	69.4	5.2
½-in. Hairfelt	59.7	59.7	6.0
¼-in. Paper-lined hairfelt	35.2	35.2	14.7
¼-in. Cabot Quilt single ply	34.3	34.3	15.6
⅛-in. Building paper	12.7	12.7	32.3
½-in. Flax board	6.5	6.5	26.2
¼-in. Sackett board	0.36	0.36	41.9
⅜-in. Sackett board	0.26	0.26	42.2
½-in. Sackett board	0.15	0.15	42.7

TABLE 6 (CONTINUED)

TRANSMISSION OBTAINED WITH MORE SENSITIVE RESONATOR

Material	Transmission		Reflection
¼-in. Sackett board	28.3	0.36	....
⅜-in. Sackett board	23.4	0.26	....
½-in. Sackett board	9.4	0.15	....
2-in. Mineral wool	18.5	0.22	16.9
2-in. Gypsum furring strips	6.1	0.073	45.8
PLASTER PANELS			
Lime base, gypsum finish	5.2	0.062	45.2
Sanded gypsum base, lime finish	6.5	0.078	45.1
Sanded gypsum base, gypsum finish	3.5	0.042	45.8
Wood fiber base, gypsum finish	3.0	0.036	46.0
Wood fiber base, lime finish	1.8	0.022	46.1

Since the materials in this series of tests were compared under circumstances maintained as uniform as possible, the results obtained are valuable for guidance when selecting material and constructions for sound-proofing purposes.

VIII. MISCELLANEOUS TRANSMISSION TESTS

33. *Test Rooms and Apparatus.*—The apparatus and arrangements for these tests were quite similar to those used in the previous ones. (See Fig. 11.) The apparatus was rearranged for convenience

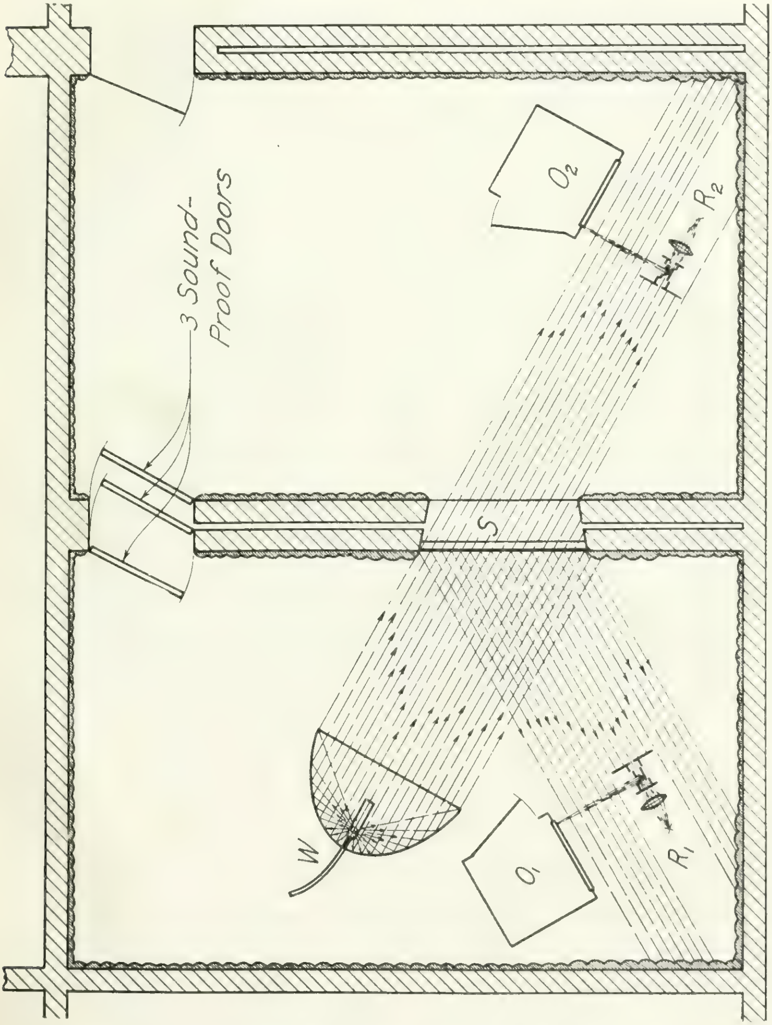

FIG. 11. ARRANGEMENT OF TEST ROOMS AND APPARATUS

and to guard the sensitive resonators from disturbance when workmen were about. The source of sound was an adjustable metal organ pipe designed for the tests and arranged to give 512 vibrations per second, this being considered an average pitch for sounds ordinarily met with in buildings. The organ pipe was blown by air from a constant pressure tank so that throughout the tests the pitch was maintained at 512 vibrations per second, with a maximum variation of 3 vibrations per second as shown by comparison with a standard tuning fork at different times. A double Rayleigh resonator developed to a high degree of sensitivity served to measure the faint sounds transmitted by the thicker wall. (See Fig. 5.) It gave responses for sounds that could barely be detected by the ear, thus furnishing a desirable instrumental substitute for the hearing, because the ear is untrustworthy in its quantitative comparison of sounds of different intensity.

34. *Further Details of Arrangements.* The amount of sound transmitted was small, so special care was taken to minimize the effect of any leakage of sound. As already explained, the bulk of the sound was directed by means of a reflector against the partition under test. The reflected sound was then largely absorbed by padding, especially by additional layers hung on the walls first struck by the reflected waves. Figs. 12 and 13 show the details.

The doorway between the rooms received special attention. Two sound-proof doors were installed in the wall of the inner room, the door casing cracks being stopped by hairfelt and plaster of Paris. These proved sufficient for the preliminary tests, but when the thicker plaster partitions were tested a small transmission of sound became noticeable. A third sound-proof door was therefore mounted in the second brick wall, after which no leakage of sound through the door could be detected by the ear.

The sound transmitted by the test partition passed directly to the Rayleigh resonator, where it produced its effect, and after this it was absorbed by padding as in the other room. Investigation with the ear verified the actions just described. It showed that an intense sound was directed to the partition and that it was then reflected strongly, the paths of the waves being as shown in Fig. 11.

Since the measurements were to be comparative, the effort was made, as in the previous tests, to maintain constant conditions. Articles of furniture and draperies in the rooms were kept fixed in position so that the interference pattern would remain unchanged. The observer placed himself in a small booth where he could take

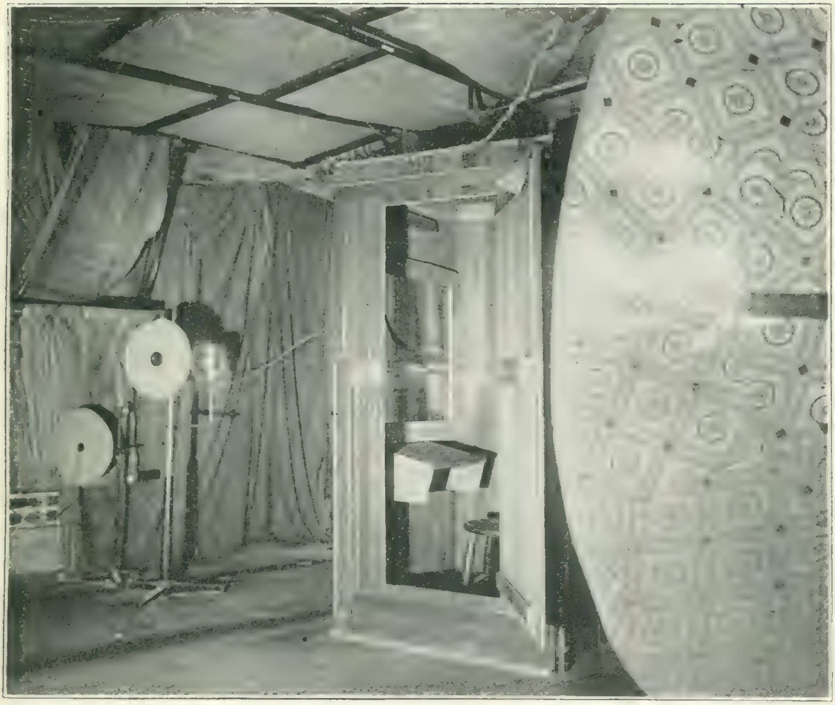

FIG. 12. VIEW OF TEST ROOM, SHOWING RAYLEIGH RESONATOR AND BOOTH FOR OBSERVER

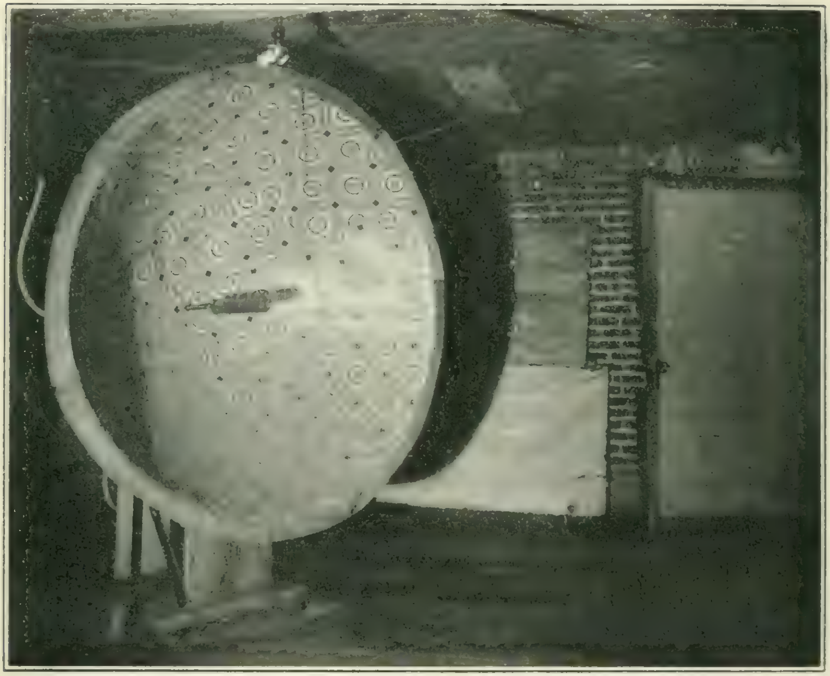

FIG. 13. VIEW OF TEST ROOM, SHOWING PARABOLIC REFLECTOR LINED WITH LINOLEUM

readings through a glass window without disturbing the action of the sound. After the initial adjustments the organ pipe and resonator were not modified. The pressure tank and the pipe connections were set up permanently at the beginning of the experiments. Changes were made only when one partition was taken out and its successor built in the wall in the same position. Where the partitions were of equal thickness the measurements were strictly comparable. The tests were conducted as rapidly as was consistent with accuracy in order to secure fairly uniform conditions of temperature and humidity, the chief delaying factor being the time required for the plaster coats to dry. The tests on the plaster walls were made during the months of June and July, 1919.

Additional measurements were taken throughout by methods other than the one described. While these extra measurements are not considered as reliable as those given by the Rayleigh resonator, they are in the same order and thus furnish an independent check on the results.

35. *Preliminary Test on Transmission of Sound Through Outing Flannel.*—It was the intention in this test to make certain that the apparatus and method would be applicable to plaster partitions, particularly those of some thickness. It was planned also to establish if possible the law connecting the intensity of the transmitted sound with the thickness of the partition. For this latter purpose a preliminary experiment was performed on the transmission of sound through outing flannel. One, two, three, four, five, and six layers of this material were mounted in succession over a metal lath core installed in the opening between the two rooms, the transmitted sound for each case being measured by a Rayleigh resonator. The results are shown graphically in Fig. 14, where the intensities of the transmitted sound are plotted against the respective thicknesses. If this were a case of pure absorption, expressible by the equation $i = i_0 a^{-x}$, as in paragraph 7, the logarithms of the intensities plotted against the thicknesses would give a straight line. The logarithms plotted for this case are shown in Fig. 14, the small departure from a straight line being explained simply by the fact that the reflected sound was not constant, but became greater with increasing thicknesses, so that the intensity of the transmitted sound fell off more rapidly than it would by absorption alone. The success of this experiment verified the various assumptions concerning the method and apparatus and gave confidence for further tests with denser walls.

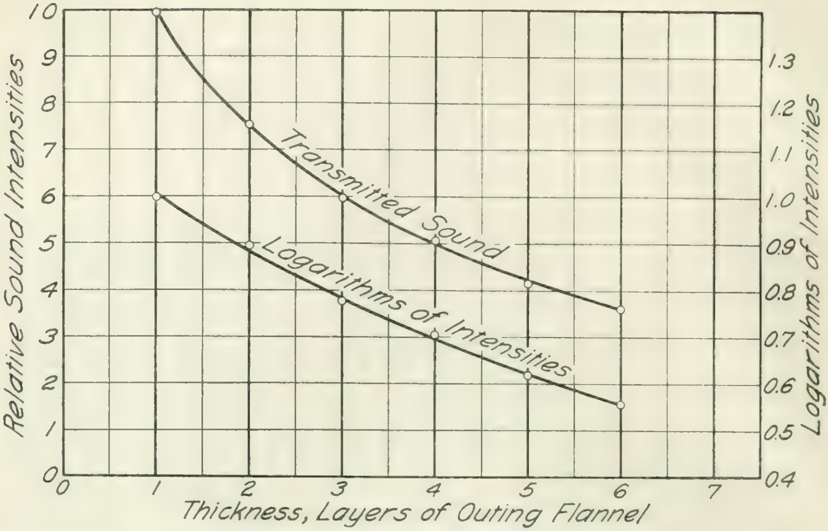

FIG. 14. TRANSMISSION OF SOUND BY OUTING FLANNEL

36. *Transmission Through Thin Plaster Partition.*—Plaster was then applied in place of the outing flannel (see Fig. 15), and it proved to be so much more effective in stopping sound that the resonator used with outing flannel gave no readable deflection, making it necessary to install one much more sensitive—the resonator pictured in Fig. 5—before a reading could be obtained. Measurements were taken on this panel for a week, until the plaster dried. More plaster was then applied with a brush to the back of the panel in order to fill the holes between the keys, thus producing a fairly uniform thickness of about ½ in. For this increased thickness the resonator gave a deflection about one-sixth as great as for the thinner layer. Further measurements were then taken for nine successive days, the results being shown graphically in Fig. 16, which displays some interesting features. The transmission increased during the second and third days after the first coat of plaster was applied, due probably to some change in the plaster as it set and dried. This phenomenon is shown for the second coat of plaster also.

37. *Effect on Transmission of Sound When Pressure is Applied to a Thin Partition.*—Finally, the transmission of sound was measured when this panel was subjected to successively increasing pressures until it broke. The pressures were applied by a 2-in. board and two

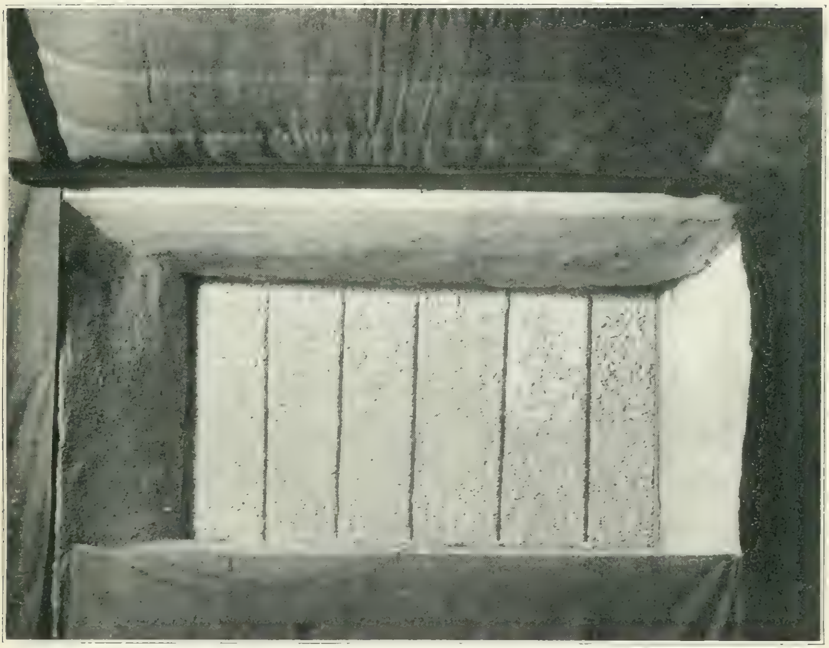

Fig. 15. Experimental Partition Formed by a Coat of Gypsum Plaster on Metal Lath

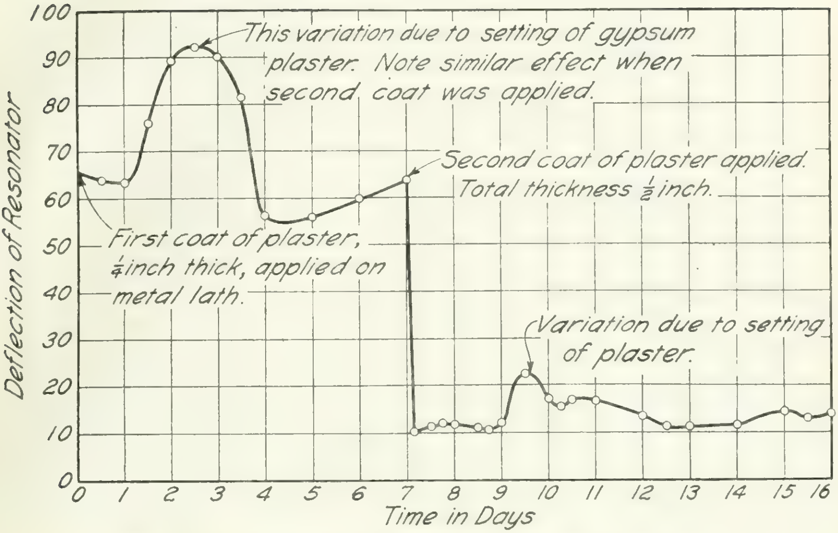

FIG. 16. TRANSMISSION OF SOUND BY PLASTER PARTITIONS OF TWO DIFFERENT THICKNESSES

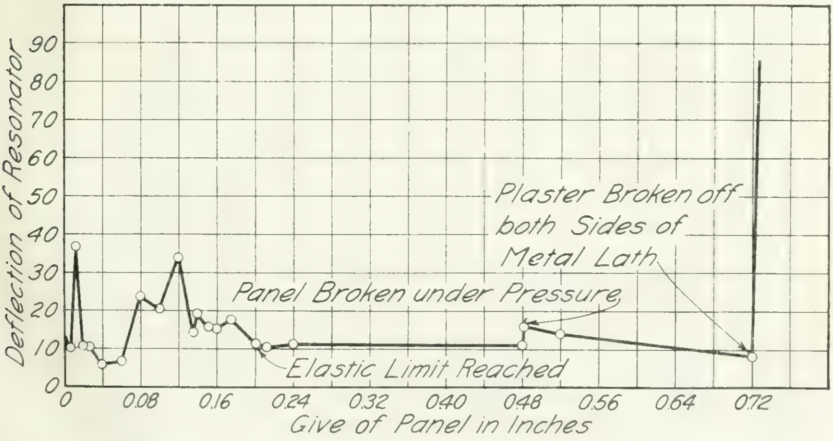

FIG. 17. TRANSMISSION OF SOUND BY PLASTER PARTITION UNDER PRESSURE

iron bars. Turning the nuts on the bolts supporting the iron bars gave successive increases in pressure. Under this treatment the panel acted very much like a drumhead, transmitting more sound when tuned up and less when something in the panel gave way and relieved the tension. Additional pressure tightened it again and the transmitted sound was increased, etc., until finally it passed the elastic limit and gave way like putty under the increases of pressure. No great amount of sound was transmitted in any case until plaster was knocked off both sides of the metal lath, thus leaving an open passage about 6 in. long. Fig. 17 gives the details.

38. *Effect of Pressure on a Thick Plaster Partition.*—The foregoing experiment was repeated with a 2-in. solid plaster board and plaster partition. A 4-in. wooden block was pushed with successively increasing pressures against the center of the partition and measurements of transmission taken at successive stages until cracks of some size were developed. The amounts of sound transmitted were not large, probably because of the friction between the rough walls of the cracks and the vibrating air particles. Figs. 18 and 19 show photographs of the partition, while Fig. 20 gives the transmission measurements.

39. *Transmission of Sound Through Threshold Apertures.*—In order to test the transmission through threshold apertures, a door was built into a 2-in. solid metal lath and plaster partition as shown in Fig. 21. The door was constructed of 2-in. wooden planks and was carefully fitted into the opening. Door stops were used on the side opposite the source of sound except at the threshold aperture. The relative intensities of sound calculated from the resonator readings are as follows:

2 in. metal lath partition, before insertion of door	0.93
With door installed with $\frac{3}{16}$-in. threshold opening	7.3
With door installed with ½-in. threshold opening	11.7

While the results probably do not express exactly the intensities of the transmitted sound because of the diffraction effects by the narrow opening, they give relative values of the effect of threshold openings. These openings are equivalent to wide cracks.

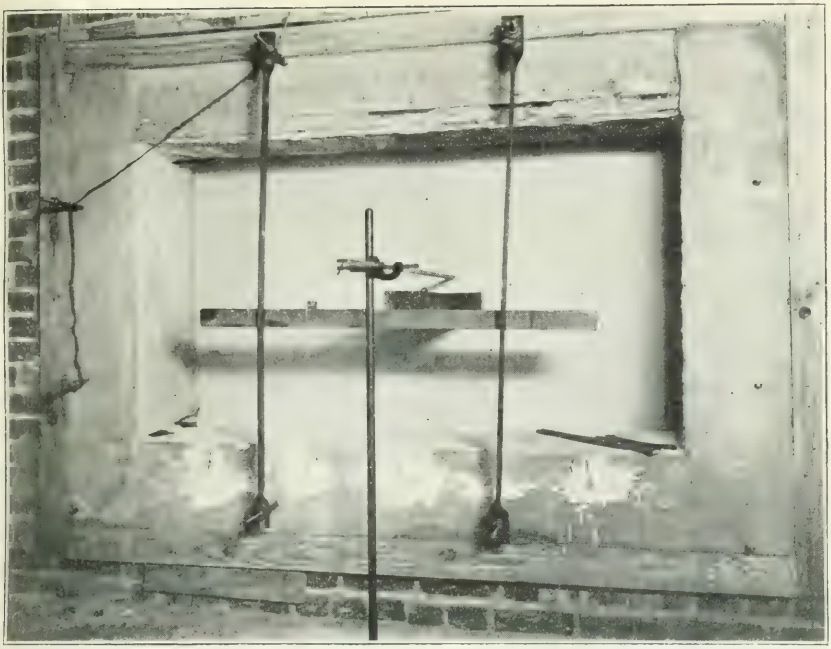

Fig. 18. Method of Applying Pressure to Partitions

Fig. 19. Cracks Developed in Solid Plaster Board and Plaster Partition

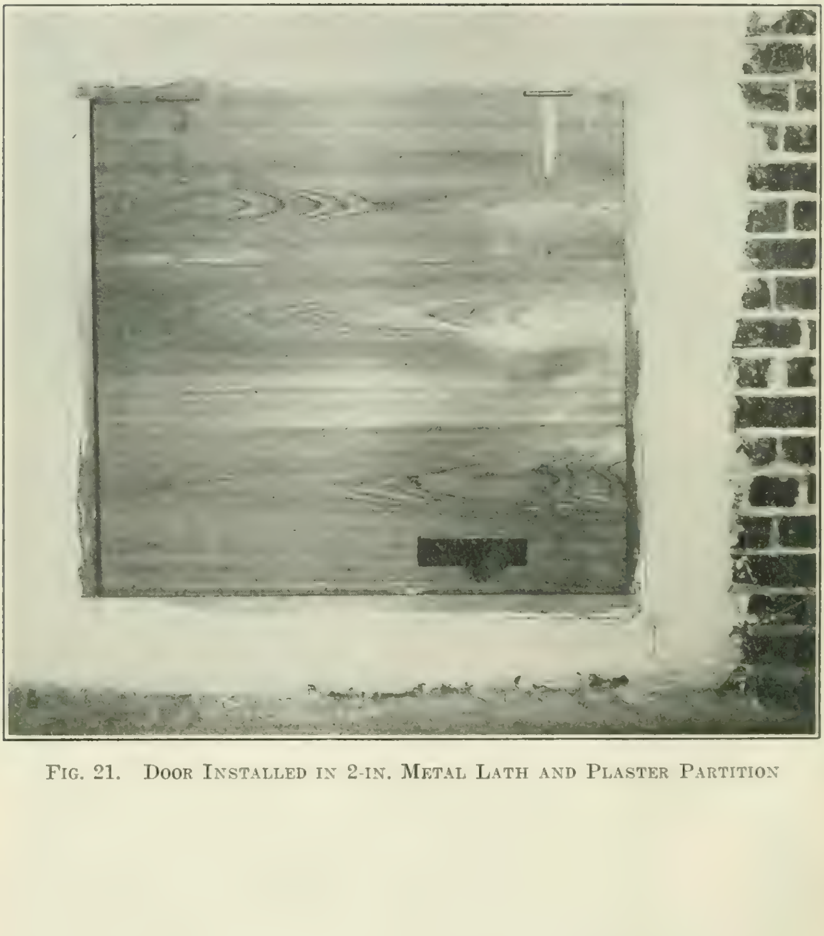

FIG. 21. DOOR INSTALLED IN 2-IN. METAL LATH AND PLASTER PARTITION

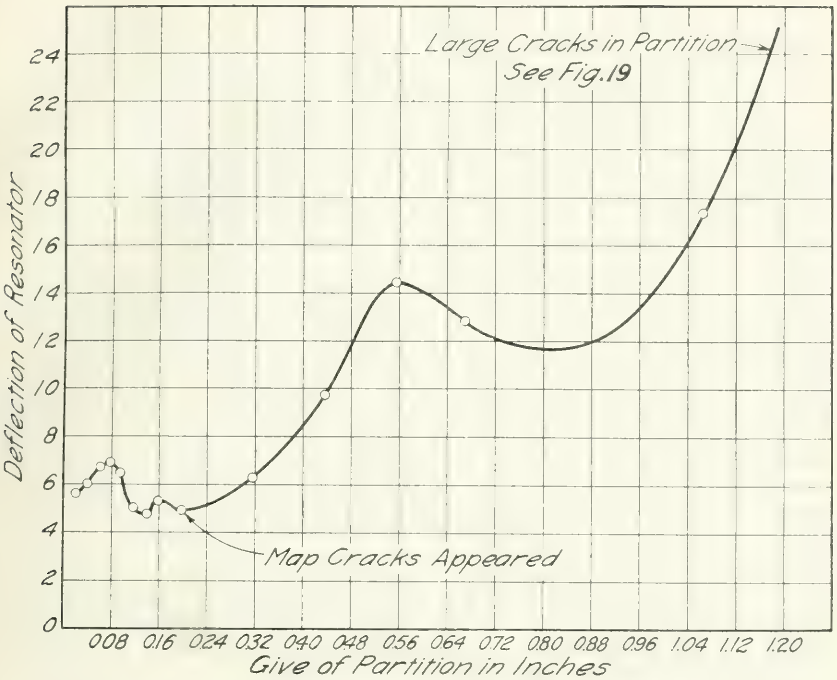

FIG. 20. TRANSMISSION OF SOUND BY PLASTER PARTITION WHEN CRACKS DEVELOP

IX. Transmission of Sound by Solid Plaster Partitions

40. *Introductory Statement.*—The preceding tests showed that the method could be applied to solid plaster partitions as well as to outing flannel and thin partitions, and that the instruments were sensitive enough to measure the fainter sounds transmitted by thick plaster. The way was now open to test thicker plaster partitions. According to the method described comparative measurements on the intensity of transmitted sound were made on the following constructions:

2-in. solid metal lath and plaster partition;
2-in. plaster board and plaster partition;
3-in. plaster block partition plastered on both sides;
3-in. plaster block partition plastered on both sides with the air holes in the plaster block filled with plaster.

The partitions, 49 in. by 64 in. in area, were erected by an expert journeyman plasterer according to the usual practical methods. All materials were purchased by a local contractor in the open market. The same kind of plaster was used on all partitions.

41. *Case for 2-in. Solid Metal Lath and Plaster Partition.*—The partitions were built solidly into the wall nearest the source of sound, the plaster coats being applied successively according to standard specifications. Instead of waiting until the partition was completed, measurements on transmission of sound were taken for each thickness, thus giving some idea of the effect of increasing thicknesses of plaster in stopping sound. Table 7 gives details of results for the metal lath partition and indicates the procedure followed for the other partitions tested for which only the essential records are shown. Each measurement recorded is the average of two measurements in most cases, in others of three.

Fig. 23, Curve 1, shows the deflections of Fig. 22 plotted against the increasing thicknesses of the partition. The transmitted sound decreases rapidly until the thickness of the wall increases to about ¾ in., after which the decrease is slower. The thicknesses for the various layers were measured when the partition was torn down, average values of several broken pieces being taken.

TABLE 7

READINGS OF RAYLEIGH RESONATOR FOR SOUND TRANSMITTED BY A 2-IN. METAL LATH AND PLASTER PARTITION*

Date	Hour	Conditions
June 24	A.M. 8.00	First coat of plaster applied on metal lath
	P.M. 2.00	72.5 cm. deflection of resonator
	4.30	68.0 cm.
June 25	A.M. 8.30	87.7 cm.
	11.30	89.7 cm.
	P.M. 1.00	second coat of plaster applied
	3.00	10.2 cm.
	4.00	9.4 cm.
June 26	A.M. 8.30	14.6 cm.
	11.30	16.2 cm.
	P.M. 1.00	third coat of plaster applied
	2.30	7.3 cm.
	8.30	5.4 cm.
June 27	A.M. 8.30	4.3 cm., drying fans put on
	P.M. 1.00	both finish coats applied
	3.00	3.8 cm., transmitted sound very faint
	8.00	3.6 cm.
June 28	A.M. 8.30	3.0 cm.
	11.30	2.8 cm.
	P.M. 4.30	1.2 cm.
June 29	1.30	3.4 cm., Sunday, quiet
June 30	A.M. 8.30	2.2 cm.
	P.M. 1.00	2.6 cm.
	5.00	2.6 cm.
July 1	A.M. 8.30	3.6 cm.
	P.M. 1.00	3.8 cm.
	7.30	3.7 cm.
July 2	A.M. 8.30	3.4 cm.
	P.M. 5.00	3.6 cm., plaster wet on surface
July 3	A.M. 8.30	3.2 cm., drying fans on
	P.M. 4.30	3.9 cm.
July 4	A.M. 8.30	3.4 cm.
	P.M. 8.00	3.5 cm.
July 5	A.M. 8.30	3.4 cm., partition dry except one spot
	P.M. 5.30	3.6 cm.
July 6	1.30	3.6 cm.

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*The data for the metal lath partition are shown graphically in Figure 22, where the deflections of the resonator due to the transmitted sound are given for successive days as additional coats of plaster were applied.

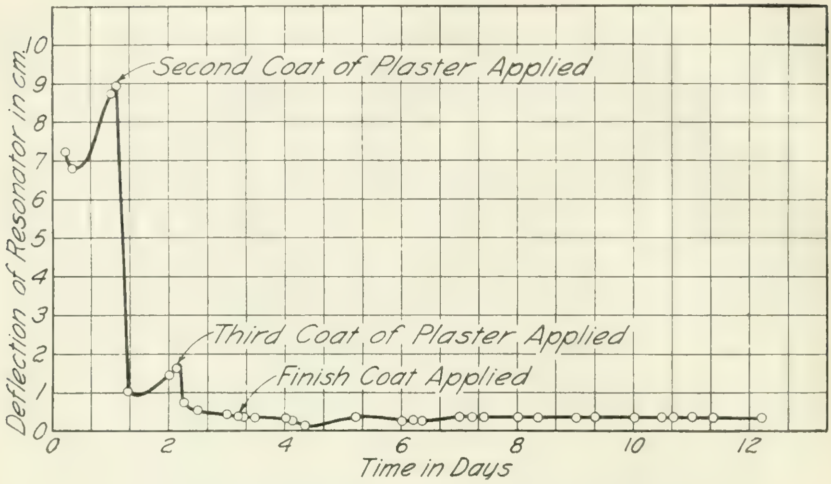

FIG. 22. TRANSMISSION OF SOUND BY PLASTER PARTITION AS SUCCESSIVE COATS OF PLASTER ARE APPLIED

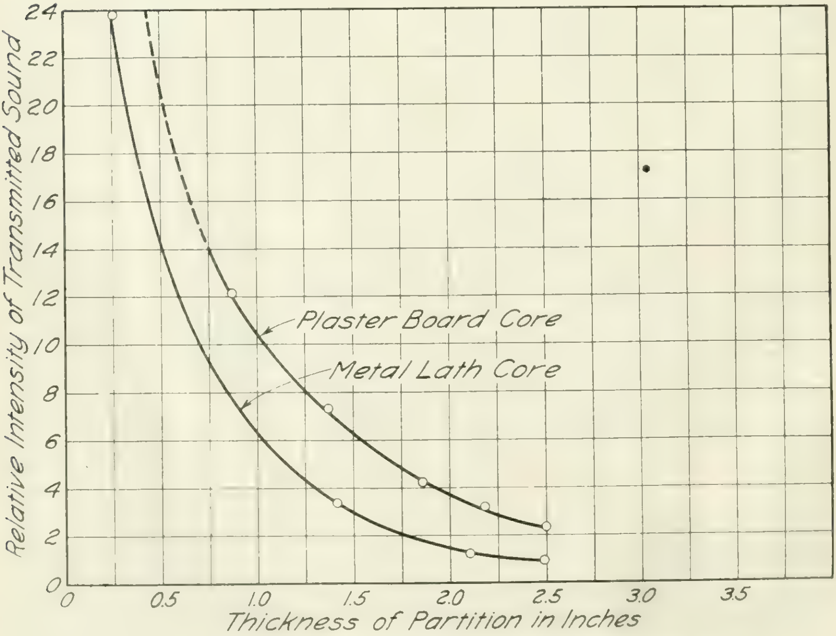

FIG. 23. EFFECT OF INCREASING THICKNESS OF PARTITION ON TRANSMISSION OF SOUND

42. *Comparative Tests of 2-in. Solid Plaster Partitions.*—A similar set of readings was taken for a 2-in. plaster partition with plaster board core. (See Fig. 23). The comparative efficiencies of the finished partitions are given in Table 8, the deflections of the resonator noted being the average of all the measurements taken after the finish coats were applied, and the intensities of the transmitted sound being calculated by using the relation given in paragraph 22.

TABLE 8

TRANSMISSION OF SOUND BY SOLID PLASTER PARTITIONS

	Average Deflection	Relative Transmission
2-in. solid metal-lath and plaster partition	3.35 cm	0.93
2-in. plaster-board and plaster partition	8.52 cm.	2.35

43. *Comparative Transmission of Sound by 3-in. Plaster Block Partitions.*—The partitions tested are described as follows:

(a) 3-in. plaster block partition plastered on both sides, giving a total thickness of 4 in.

(b) 3-in. plaster block partition plastered on both sides with the air holes in the plaster blocks filled with plaster, giving a total thickness of 4 in.

The measurements were taken in the same manner as for the 2-in. partitions. The plaster blocks for test (b) were filled with plaster and allowed to dry before erection into the partition. The results obtained are as follows:

	Average Deflection	Relative Transmission
Partition (a)	13.9 cm	3.85
Partition (b)	4.23 cm	1.16

44. *Effect of a Rigid Plaster Partition on Sound.*—The transmission of sound through a plaster partition appears to depend on its rigidity and mass. Thin partitions transmit considerably more sound than thick ones, largely because they are less rigid, and vibrate more easily. Vibrations are set up which may become quite large when the natural frequency of the partition is in tune with the incident sound. Thick partitions on the other hand are more rigid and vibrate less, so that they stop sound largely in proportion to their mass.

Sound in a room may be transmitted through the partitions with more or less difficulty, depending on the qualities of the structure. Thus, the *inertia* of a partition plays an important part. On striking the heavy partition particles, the air particles of small weight are thrown back in much the same way that a tennis ball would be, on striking a cannon ball. Fig. 24 pictures a conception of this idea.

The partition particles are moved back and forth slightly by the incident pressures and rarefactions, so that sound waves greatly diminished progress through the partition. Since the latter is thin compared with the wave length of the sound, all the particles in it may be thought of as moving simultaneously in the same direction; that is, the partition may be considered to move back and forth as a unit. This motion is carried to the air on the further side where diminished waves, constituting a faint sound, are set up.

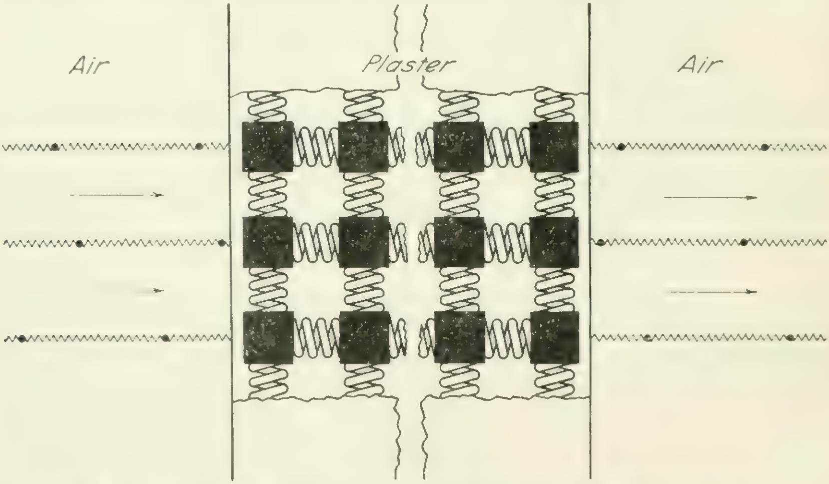

FIG. 24. DIAGRAM ILLUSTRATING TRANSMISSION AND REFLECTION OF SOUND WAVES BY A SOLID PARTITION

Theory shows that the sound in air reflected from a rigid material is given by the expression:*

$$-\frac{\pi \rho_1 l/p \lambda}{\sqrt{1+\pi^2(\rho_1 l/p \lambda)^2}}$$

where ρ is the density of air, ρ_1 that of the material, λ is the wave length of sound, and l the thickness of the material. Calculations from this expression show that the sound reflected from a rigid gypsum wall 2.5 in. thick is 99.99968 per cent where the incident sound is taken equal to 100 per cent. The transmitted sound is the difference between these quantities, or 0.00032 per cent, giving approximately a transmission of 3 parts in 1 000 000 assuming that no sound is absorbed in passing through the wall or lost in any other manner. The determining factor in the reflection is the ratio of the density of air compared to that of plaster; that is, the wall behaves like a rigid body and acts mainly because of its inertia. It is quite unlikely, however, that such a partition will be absolutely rigid, particularly if it has some area; therefore, some sound will be transmitted because of vibrations.

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* Rayleigh, Lord. "Theory of Sound." Vol. 2, Sec. 271, Equation 11.

45. *Effect of a Flexible Plaster Partition on Sound.*—In case the partition is limited in extent and fastened at the edges, as it must be in any form of partition construction, the sound pressures must overcome not only the inertia of the plaster but also its *rigidity* or resistance to being distorted. A sound pressure applied perpendicularly to the surface causes a small displacement, greatest at the center and zero at the edges, so that the partition becomes slightly bulged, with a small increase in area. The less a wall gives under such a force, the more efficiently will it stop sound. An inspection of Figs. 24 and 25 in this connection give some idea of the strains set up by a

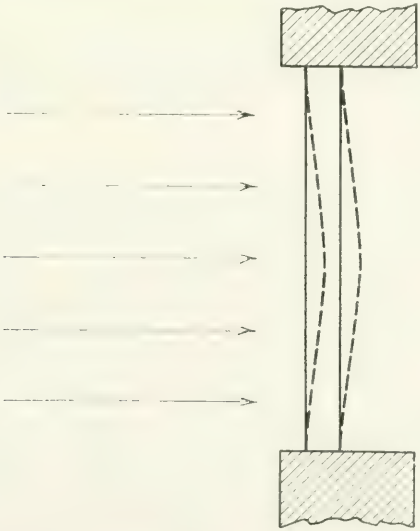

FIG. 25. HOW A PARTITION IS BULGED BY PRESSURE OF SOUND

bulging force. It should be remembered that the pressures due to sound are not static but are rapidly alternating pressures and rarefactions of very small amplitude.

The transmission of sound depends on the vibration of the partition, which acts in this regard like a thick elastic plate. The rapidly alternating pressures and rarefactions of the sound waves set the partition in minute motion and thus create corresponding pressures and rarefactions (sound waves) on the farther side of the partition. The effect is intensified and the transmitted sound increased in volume if the natural period of the partition is in tune with the sound waves. The vibrations for ordinary sounds and partitions are usually small, probably not exceeding 1/100 in. in amplitude. Other factors being the same, they decrease in intensity in walls of greater rigidity.

For a rectangular partition fixed at the edges the period of vibration, N, is deduced from the equation:*

$$N = \frac{\pi h}{2}\left(\frac{m^2}{a^2} + \frac{n^2}{b^2}\right)\sqrt{\frac{q}{3\rho\,(1-\sigma^2)}}$$

where q is Young's modulus of elasiticity, ρ the density of the partition, σ is Poisson's ratio of lateral contraction to longitudinal extension, a, b, and $2h$ the length, breadth, and thickness of the partition, and m and n whole numbers depending on the mode of vibration. This equation shows that thick elastic partitions vibrate quickly, but massive partitions of some dimensions vibrate more slowly.

46. *Effect of the Structure of Plaster Partition.*—Another factor affecting the transmission of sound through a partition is the character of the structure. Compared with a thin partition a thick, homogeneous structure has the advantages of greater inertia and rigidity. The use of an air space completely separating two members of a rigid, non-vibrating double partition would have a marked action on sound, and, according to theory, a partition of this construction would stop many times more sound than a similar single partition whose thickness equals the sum of the thicknesses of the two members of the double partition. This is due to the abrupt change in elasticity and density

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* Barton, E. H. "Text Book of Sound" Sec 226 Equation 11
Rayleigh, Lord "Theory of Sound," Vol. 1, Sec 225

from plaster to air and from air to plaster as the sound strikes the second member. In case the air space is bridged over, as in practical constructions is usually the case at the ceiling, floor, and other points, this theoretical efficiency is greatly diminished because the vibrations travel easily along the paths afforded by the continuity of solid materials. Thus the bridged over partition should be considered as a unit instead of two separate members, and its efficiency in stopping sound judged mainly on its weight and rigidity.

The core of a partition is another feature of structure that affects the transmission. It may be of such a nature as to increase the strength of the partition, it may be simply the central part of a homogeneous medium, or it may so separate the partition into two parts that the structure is weaker than a homogeneous unit. A partition with increased strength due to the core, such as a steel reenforcement, would be more rigid than an equally thick homogeneous partition and would stop more sound. The homogeneous partition in turn would be more efficient in stopping sound than the double partition weakened by the core. The latter, however, has some possible advantage in reflecting sound because of the change in elasticity and density in the core. An extreme illustration of this kind would be the case where the core consists of hairfelt, so that its action in stopping sound would be analogous to that of an air space. If the core consists of a sheet of thick paper, making, without air space, a continuous contact with plaster on both sides, this efficiency is largely lost and the small gain due to reflected sound would appear to be overcome by the loss in rigidity of the structure.

47. *Discussion of Experimental Results Obtained with Solid Plaster Partitions.*—The experimental results obtained in the tests appear to be in accord with these theoretical considerations. Thin partitions vibrated under the action of sound waves, as could be ascertained by touching the surface of the plaster. The vibrations (and transmitted sound) were more vigorous for the proper coordination of the elasticity of the partition and the frequency of the sound as shown by the data pictured in Fig. 17. Thick partitions transmitted but little sound. They were quite rigid because of their small area and considerable thickness, and also because they were mounted solidly in a double brick wall. Their action in stopping sound must have been due largely to their rigidity and mass.

It should not be concluded from these tests that partitions of similar construction in buildings will all have exactly the same sound-proof qualities; larger area of building partitions, resulting in lessened rigidity, will allow a greater transmission of sound. The value of the results obtained in the tests lies in the fact that they were obtained by direct comparison under identical surroundings, rather than by tests on different types of partitions in different buildings, with varying floor and ceiling constructions, unequal sizes of rooms, uncontrolled extraneous sounds, etc. The results thus give information for guidance in the choice of materials and constructions where sound insulation is contemplated.

X. Examples of Sound-Proof Rooms

48. *Preliminary Statement.*—In order to supplement the information given in the previously described experiments examples of soundproofing should be included in the discussion. These examples are mostly practical constructions in buildings with walls of greater size than could be tested conveniently in laboratory experiments, and with additional complexities such as a ventilation system, pipes for steam, water, and wires, etc. An account of such constructions, together with the results of the previous tests, allows more definite conclusions to be drawn concerning the general procedure in sound insulation.

The illustrations chosen for discussion concern two types of installations; first, single rooms that are soundproofed for experimental purposes, and second, entire buildings where sound insulation was needed. A detailed description of all the examples available would involve too long an account; therefore, several typical cases are selected for discussion and references are given for the others.

49. *Sound-Proof Rooms for Acoustic Experiments.*—A small laboratory building containing a number of rooms for acoustical experiments was designed by W. C. Sabine according to his extensive experience in architectural acoustics.* The essential construction was simple yet effective. Heavy brick walls served to give sufficient rigidity to reduce vibrations to a minimum. The weight of these walls together with an air space between them presented a decided hindrance to the transmission of sound waves. (See Fig. 26.)

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* Sabine, P. E. "Wallace Clement Sabine Laboratory of Acoustics." Am. Arch., Vol. 116, pp. 133-138, 1919.

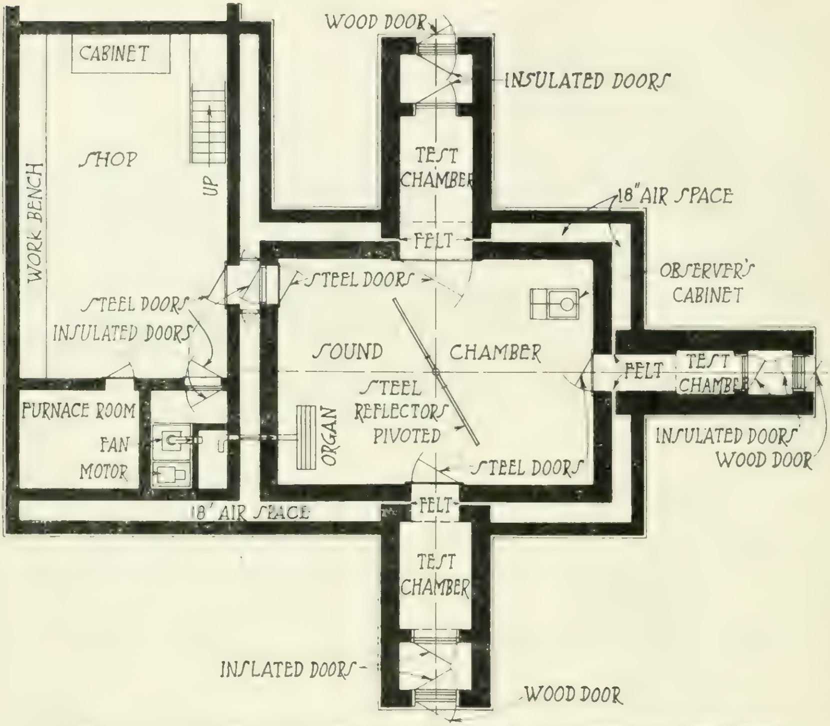

FIG. 26. PLAN OF WALLACE CLEMENT SABINE ACOUSTICAL LABORATORY

(This laboratory was built and is maintained by Colonel George Fabyan at Geneva, Illinois, because of his interest in scientific work and his personal regard for Professor Sabine. Since the death of the latter the investigational work has been conducted by Dr Paul E. Sabine, a cousin of Professor Sabine.)

50. *Sound-Proof Constant-Temperature Room.*—The structure illustrated in Fig. 27 is a small building constructed primarily for securing constant temperature. It appears also to be an excellent sound-proof structure. The description* is as follows:

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* "Results of Observations made at the Coast and Geodetic Survey Magnetic Observatory at Cheltenham, Maryland, 1901-1904." Dept. of Commerce and Labor, U. S. C. and G. S., pp. 11-13, 1909.

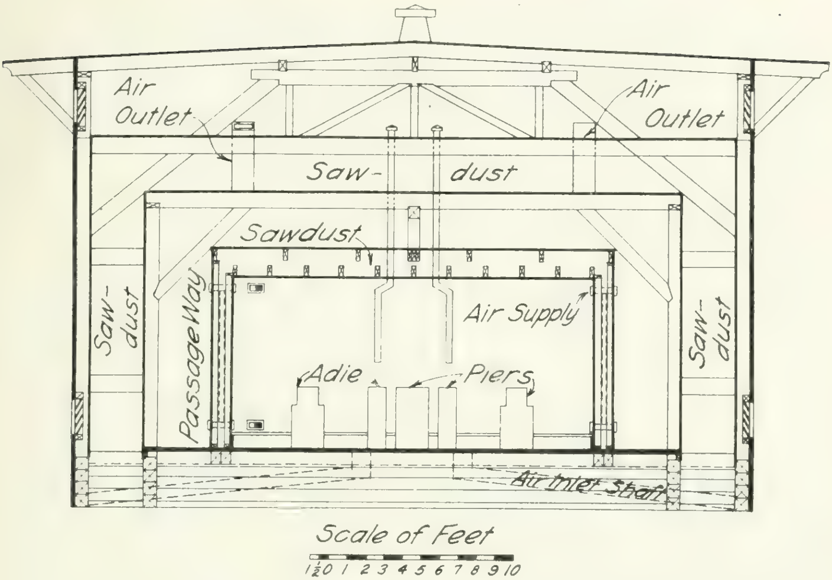

FIG. 27. VERTICAL CROSS-SECTION OF SOUND-PROOF CONSTANT-TEMPERATURE ROOM

"The variation observatory is essentially two small buildings inclosed by a third larger one. The outside dimensions of the variation observatory are 36 by 56 by 24 ft., with a vestibule 10 by 13 ft. The inner rooms are each 16 by 19½ ft. inside measure, separated from each other by a passageway 5½ ft. wide. The wall insulation is as follows (see Fig. 27); beginning at the outside, pine weatherboarding, 8-ply building paper, 1-in. pine sheathing, 8-in. air space, 1-in. pine sheathing, 8-ply paper, 3 ft. pine sawdust, 8-ply paper, ⅞-in. pine sheathing, 3 1/16 ft. air space (passageway around inner rooms), ⅞-in. pine sheathing, 8-ply paper, 1 ft. pine sawdust, 8-ply paper, ⅞-in. pine sheathing. Beginning at the roof and going down: gravel and asphalt-pitch roof, 1-in. pine sheathing, 3 2/3 ft. air space, 1-in. rough pine floor, 3 ft. pine sawdust, 8-ply paper, ⅞-in. pine sheathing, 3 ft. air space above inner rooms, 1-in. rough pine floor, 1½ ft. pine sawdust, 9-ply paper, ⅞-in. ceiling. Insulation from bottom of foundation is 2 2/3 ft. earth, 6 to 8 in. layer of gravel, 3 ft. sawdust, 1-in. pine floor, ⅞-in. matched pine floor. The 8-in. air space next to the outside of the building is provided with slat ventilators at top and bottom, which when open permit a free circulation of air up the sides of the building and out through the ventilators in the roof. By the use of tight-fitting shutters it can be converted into a practically air-tight space. By this arrangement of alternating air spaces and sawdust packing the variation in the temperature is kept within the desired limits without the aid of heating apparatus. The daily range is kept down to almost nothing and the annual range to only a few degrees Centigrade."

Mr. W. W. Merrymon, one of the magnetic observers, stated that the room was so sound-proof that he thought he would be unable to make outsiders hear in case he were in distress —from lightning or other causes. Trial by loud shouting within the room gave almost no sound outside, and this faint sound appeared due to the passage of sound through the air ducts in the floor. By closing these the room would doubtless have been made "sound-proof."

51. *Sound-Proof Room for Psychology Tests.*—Concerning this room, Dean C. E. Seashore of the State University of Iowa writes: "Our experience in building the sound-proof room in psychology would not be of any great help in the soundproofing of a music building, since all the precautions that we have taken are such as could be taken for a small cage room." The description* of the room is as follows:

"The construction of the observing room deserves especial mention. To make a dark room impervious to external light is a matter presenting no serious difficulty. To make a room impervious to external sound or wholly free from the jarring from surrounding rooms or adjacent streets is a problem which has not yet been solved and of course never will be. We made the attempt to approach a little nearer to this end than has hitherto been done. The result is a room as free from external disturbances as is needed in any experiments in which it is necessary to control visual, auditory and tactual stimuli. So far as this has been accomplished, the credit is largely due to the architects, Messrs. Proudfoot and Bird, who worked out many of the details of construction. The position of the observing room is central, occupying a place not otherwise desirable from lack of light. The room rests on an independent foundation, having no solid connection with the rest of the building either below, above, or on the sides. The superstructure which supports the room rests upon a sand bed and a second sand bed at a higher level still further assists in eliminating possible jarring or sound which might be communicated from the ground. The walls of the room itself, inside the main partitions, which separate the whole space from the surrounding apartments, are made of two four-inch walls of hollow tiles separated by an air space and each covered with a thick insulating material made of sea-weed. Inside of all, the walls are plastered and then lined throughout with black broadcloth. The inside room is divided into the main observing room 12 ft. 2 in. by 12 ft. 7 in., and a vestibule, 4 ft. by 12 ft. 7 in. The room is entered through five doors, the outer one being an ordinary oak door and the other four especially constructed tight fitting cedar doors, covered with black cloth on the sides and edges. The doors close with strong springs and are held open by automatic catches. The floor is made of Tennessee red cedar and covered with linoleum painted black. The room is heated and ventilated by means of hot air introduced not directly but from the attic through cedar-shafts provided with overlapping cloth partitions which admit the air but help to exclude sound. For very fine experiments the ventilating shaft may be closed and the room ventilated during intermissions. The room may be lighted by gas or electricity, the former being introduced through rubber tubes. The furnishing consists of black tables and chairs and a telephone connected with the measuring room. In the exclusion of sound and vibration, as well as in other respects, the room has proved to be a complete success. The loudest stentorian shouting just outside the doors is absolutely unheard within.''

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* Studies in Psychology, Vol. III, pp. 132-143; State Univ. of Ia., Bul., New Series, No. 49, May, 1902.

52. *A Noiseless Room.*—A noiseless room for sound experiments was built in Utrecht, Holland.[*] Figs. 28 and 29 show the complicated details. The room was about 7 ft. by 7 ft. by 7 ft., and the walls were 11 in. thick. A person in this room could hear his heart beat.

53. *A Group of Sound-Proof Rooms.*—Norton[†] tested five small rooms whose walls were specially constructed of different materials. The tests were not strictly comparative because three of the rooms had one thin side that vibrated under resonance and transmitted sound to adjacent sides; also there was some leakage of sound through the doors and ceiling. Observations were taken by the ear, and different partitions rated by estimation. The results show that increasing the thickness of the walls promoted sound insulation; also, that double partitions with air space and enclosed sound absorbing materials are more effective than single partitions.

54. *Room Soundproofed for Machine Gun.*—During the war a room about 12 by 14 by 12 ft. was designed to insulate the explosive sounds of machine guns when these were being tested. The room occupied one corner of a larger room. The walls of the insulating room were brick on two sides—those of the larger room—and wood on the other two sides. Four inches of ground cork lined the wood walls, being held in place by metal netting nailed to 2 in. by 4 in. wood studding. A double layer of flax boards lined the ceiling. A pile of sand into which the bullets were fired assisted in the absorption of sound. People across the street from the room were unaware of the machine gun operation until the observer forgot to close the double windows after opening them to let the powder smoke out.

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[*] Franz, S. I. "A Noiseless Room for Sound Experiments." Sc., Vol. 26, pp. 878-881, 1907.

[†] Norton, C. L. "Sound-Proof Partitions." Insur. Eng., Vol. 4, p. 181, 1902.

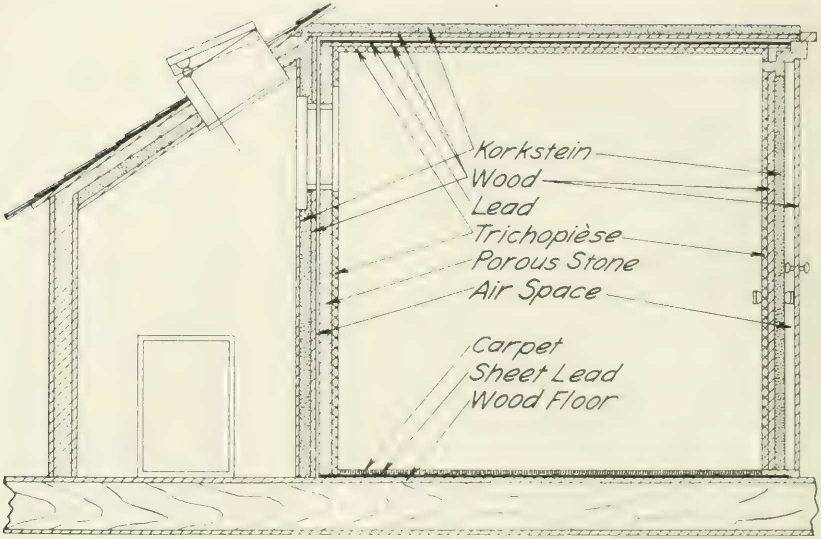

FIG. 28. VERTICAL CROSS SECTION OF NOISELESS ROOM

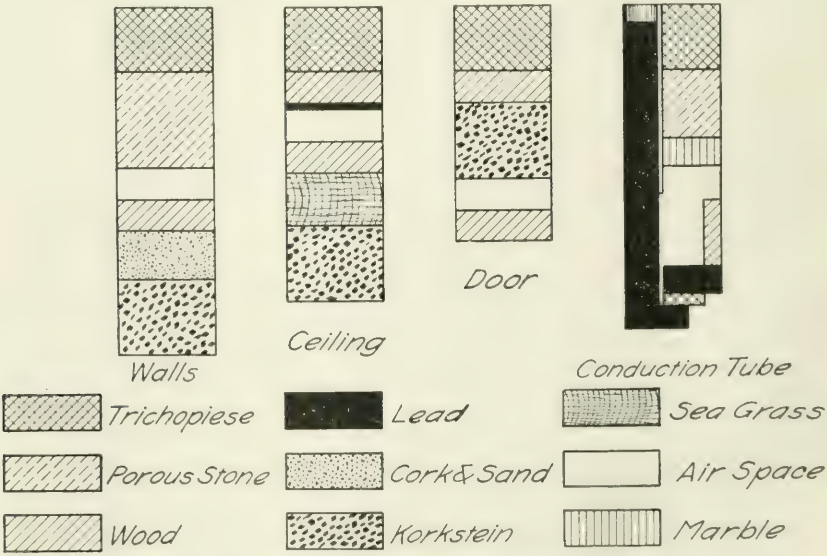

FIG. 29. DETAILS OF CONSTRUCTION OF NOISELESS ROOM

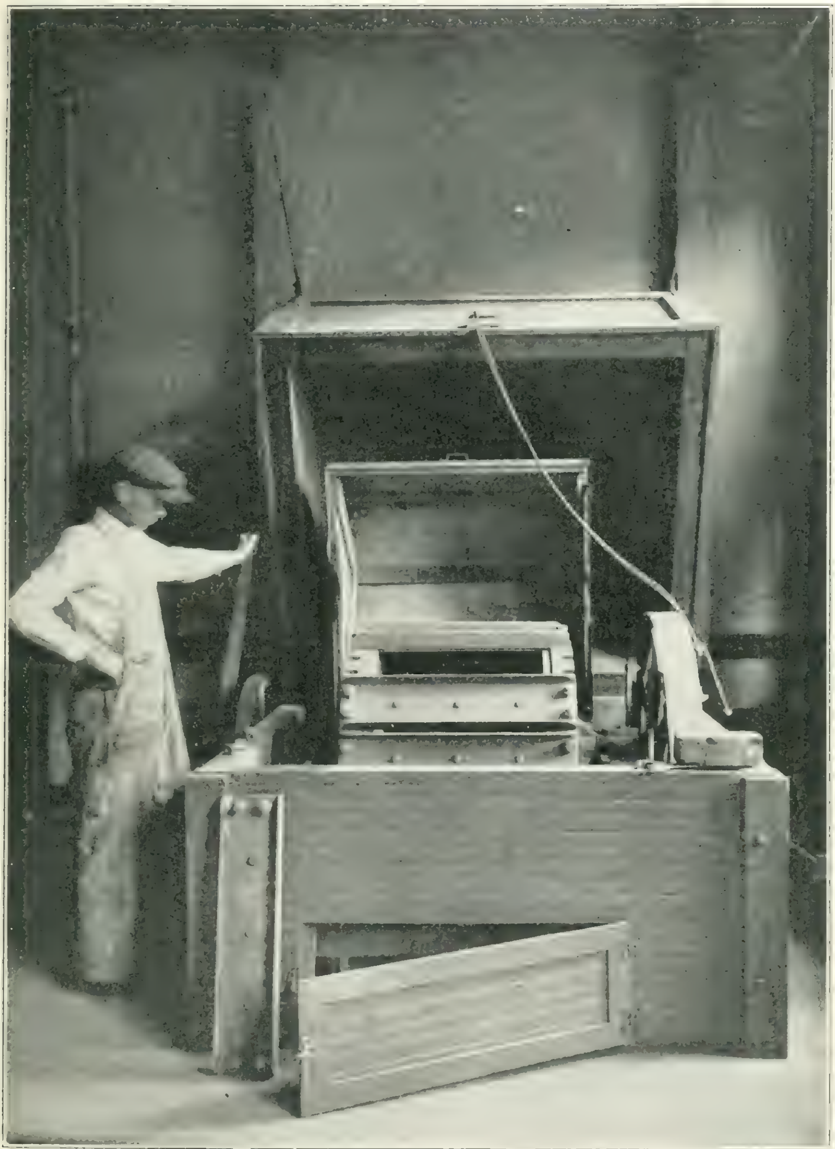

FIG. 30. SOUND-PROOF BOX TO REDUCE NOISE OF BRICK RATTLER

55. *Soundproofing a Noisy Machine.*—A brick rattling machine, used by the Highway Engineering Department at the University of Illinois, was extremely noisy, so that suggestions for a method of minimizing the disturbance were requested. The machine was mounted on a platform and enclosed first in a sheet iron covering which served to prevent the escape of dust as well as sound. (See Fig. 30.) A larger, double walled wooden box, having the interspace filled with saw-dust, and being equipped with a hinged cover, was built to give further insulation. Without the covers, the machine is so noisy that one person cannot be heard by another without shouting loudly into his ear. With the covers on, the noise is greatly reduced and conversation is possible. This matter has its economic side, because, with noisy machines, the hearing and therefore the efficiency of employees becomes affected.

XI. SOUND-PROOF BUILDINGS

56. *The Smith Music Building.*—An increasing number of buildings with soundproofing features have been constructed in the past few years. F. R. Watson, in collaboration with Architect James M. White, incorporated a number of soundproofing constructions in the Smith Memorial Music Building at the University of Illinois.[*] This problem was more complex than soundproofing a single room. It involved the sound insulation of some fifty small practice rooms, twelve studios, and the larger concert hall, besides the acoustic control of sounds of motors, fans, and elevators.

Since the possibility of transmission of sound was greatest between adjacent rooms, each dividing wall, ceiling, or floor was made double, with air space containing absorbing material, and was left entirely unbroken. All pipes, conduits, ventilator ducts, doors, and windows were specially placed in outside or corridor walls where the leakage of sound would be less harmful. This systematic construction *throughout* the building meant that a sound generated in a room must penetrate the insulation to escape. To enter another room, it must pass a second time through a special insulation. When traversing the building structure, a sound would continually meet hindrances that would either stop or absorb it.

Fig. 31 pictures some of the features that were adopted to control sound. The concrete floor, 12 in. thick, was broken in its continuity by the form planks that were purposely left in place. Walls between rooms were constructed of two 3-in. gypsum partitions insulated at the bottom by machinery cork and at the top and sides by hairfelt. "Insulite" was installed in the air space between the gypsum partitions, to absorb sound and also to present a barrier in case cracks developed in the gypsum. The finish floors were floated on a 1-in. layer of dry sand in order to break the continuity of material and thus stop the progress of vibrations.

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[*] Watson, F. R. "Sound Proofing a Building." Arch. For., Vol. 35, pp. 178-182, Nov., 1921.

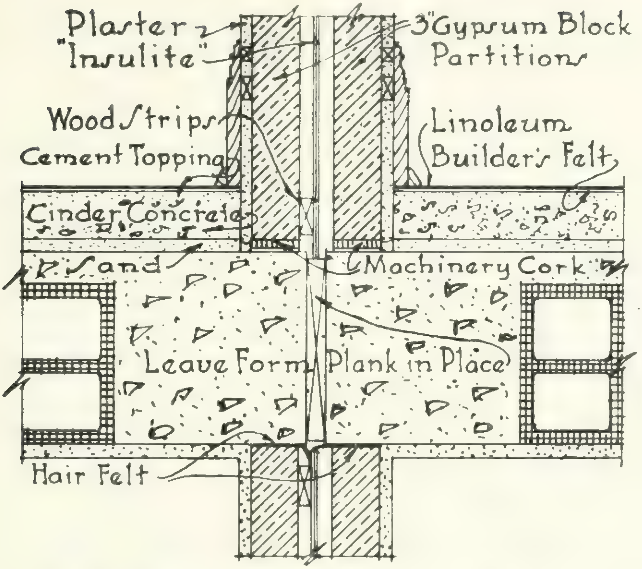

FIG. 31. DETAIL OF FLOOR AND PARTITION CONSTRUCTION IN SOUND-PROOF BUILDING

Experiments conducted in the building after its completion show that a measure of success attended the design and construction. Loud speaking and shouting in the practice rooms can hardly be heard outside. Music, however, penetrates the insulation more easily, so that sound, largely diminished, may be heard in adjacent rooms. This leakage of sound, however, does not appear of great disadvantage. Students use adjacent rooms for singing practice, and for piano, violin, and other instrumental drill, without serious disturbance to each other.

The ventilation system is not as sound-proof as desired, and appears to be the greatest drawback in the control of sound. Each room was equipped with a separate inlet and outlet duct. Four independent ventilating systems furnished air to four groups of rooms in order to lessen the chance for transfer of sound from one group to others. The ventilation system is now under investigation with a view to improving the insulation.

The building is not absolutely sound-proof nor does this appear necessary for practical purposes. The sounds that leak through the insulation are greatly diminished in intensity and therefore of minor importance compared with the sounds generated in the room containing the observer.

57. *Other Buildings with Soundproofing.*—Music buildings with soundproofing features have been erected at other institutions, and, from the reports received, appear to give satisfaction. The constructions involve double walls with absorbing material, and more or less attention has been given to the design of the ventilation.

XII. CONCLUSIONS

58. *Summary of Conclusions and Recommendations.*—The information in this bulletin was drawn from three sources; the theory of the behavior of sound waves, experimental investigations of the effect of materials on sound, and examples of sound-proof installations. The details of this information, while drawn from different sources and apparently unrelated, coördinate in a satisfactory way in setting forth similar conclusions.

Some of the more general principles and recommendations are stated in the following paragraphs, but the details and comments necessary for a more comprehensive conception of the problem of soundproofing are to be found in the descriptions throughout the bulletin.

Sound may be transmitted from one side of a partition to the other in three ways; it may progress through continuous air passages, it may pass as an elastic wave through the solid structure of the partition, or, by setting the partition in vibration, it may originate sound waves on the further side.

These actions are quite readily understood by remembering that sound consists of a series of compressions and rarefactions that progress rapidly through a medium without interruption unless they meet a new medium with a different elasticity or density. For instance, sound waves in air proceed without hindrance through air passages, such as ventilation openings in a partition. If, however, the passages are small in cross-section, as in the case of a porous material, the progress is hindered and a certain amount of absorption of the energy takes place, due to the friction set up between the vibrating air column and the sides of the pores.

In case the partition is impervious to air, the direct progress of the waves is interrupted. A thin partition is set in vibration and thus orginates new waves on the side opposite the incident sound. For a thicker, more rigid partition, the vibrations are smaller and a very considerable part of the energy is reflected. The transmission in this case takes place by compressional waves communicated to the solid material of the partition. The amount of energy thus transmitted is usually quite small.

In view of these considerations a sound-proof partition should be as rigid and free from air passages as possible. For effective soundproofing of a group of rooms, the partitions, floors, and ceilings between adjacent rooms should be made continuous and rigid. Any necessary openings for pipes, ventilators, doors, and windows should be placed in outside or corridor walls where a leakage of sound will be less objectionable.

In case the sound is generated in the building structure, as the vibrations set up by a motor fastened to the floor, the compressional waves proceed through the continuity of solid materials. In order to stop them, it is necessary to make a break in the structure so as to interpose a new medium differing in elasticity and density. For instance, the vibrations of a motor may be minimized by placing a layer of hairfelt, or similar air-filled material, between the supporting base and the floor. Where the machine is quite heavy, footings may be made of alternate layers of asbestos, lead, and leather. Bolting through this material will reduce the insulation, because the vibrations in this case will pass easily through the bolts to the floor. The insulation should thus be left without any bridging over of the discontinuities. Air gaps in masonry will be effective if the air space is not bridged over at any point. A floor floated on sand, sawdust, or hairfelt would approximate this condition. The edges of the floor should be insulated from the walls by felt or similar material.

Especial attention should be paid to the ventilation system. All effective sound-proof constructions either omit entirely a ventilation system or else construct it in some special manner to avoid transmission of sound. In some buildings air is supplied and withdrawn from rooms by individual pipes that are small in diameter and extend without break from the air supply chamber to the rooms. This results in considerable friction between the walls of the pipes and the air, with a resultant weakening of the sound waves. Without some efficient control of the transference of sound through the ventilation system it is a waste of effort to construct sound-proof walls, double doors, and other contrivances for insulation.

When soundproofing a building all details should be considered with respect to the likelihood of transmission of sound. Each room, as far as possible, should be made an insulated unit by means of air spaces or air-filled materials that separate it from surrounding walls. Pipes and ventilators should be so installed as to minimize the chance of transfer of sound. Patent doors are now available that will close the door space at top, sides, and bottom. In case a troublesome sound is generated in the room, it may be minimized by installing absorbing material on the walls.

The absorption of sound is an essential feature for soundproofing. Reflecting sound and scattering it still leaves it with energy. It must be absorbed; that is, converted into heat energy by friction, before it is eliminated as sound. This means that carpets, furniture, draperies, etc. should be present, or if greater absorption is desired, hairfelt or similar materials must be installed.

The insulation of sound is a complex problem and a successful solution is obtained only when all the possibilities of transfer of sound are anticipated and guarded against. While many things may be learned from further experience and much may be gained from additional theory, enough has been revealed to give encouragement to the belief that soundproofing may be prescribed in the future with some of the certainty that now attends the acoustic design of auditoriums.

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THE IGNITION TEMPERATURE OF COAL

BY

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THE IGNITION TEMPERATURE OF COAL

I. Introduction

1. *Outline of Work.*—There is no definite temperature at which coal bursts into flame. The phenomenon of flame from coal is due to the combustion of volatile matter driven off from the coal, the character of such matter varying under different conditions; hence, the temperature at which the coal flames varies widely and is dependent upon surrounding conditions, and therefore the flaming temperature of a coal does not serve as an indication of the ignition point. Strictly speaking, it is the true ignition point, but since it varies so widely its evaluation is meaningless. Moreover, any definition of ignition temperature based on the spontaneous emission of heat from coal is equally unsatisfactory, because experiments have proved that coal is giving off heat of combustion or some phenomenon akin to combustion at all temperatures. To be sure, the combustion at low temperatures is slow, but it is nevertheless evident. As the coal is raised in temperature by an outside source of heat it produces more and more heat from its own combustion, but there is no point in this process where the change in rate of heating is sufficiently abrupt to be regarded as indicating a definite ignition temperature.

The first purpose of this investigation, therefore, was to establish some definite point along the line of this process of heating which could be called the ignition temperature. In the effort to locate such a temperature heating curves were drawn to show the rate at which the coal increased in temperature when assisted by an outside source of heat. After a study of these curves the point at which the coal assumed a uniform glow was chosen as the most logical ignition temperature, not only because this point was found to be rather definite in the heating curve, but also because it could be checked in various types of apparatus by different means of temperature measurement and control, and by different operators.

With this glow point chosen as the ignition temperature the method for the determination became comparatively simple. The sample of coal was heated gradually in a normal air supply until the glow appeared, the temperature of the coal itself being measured by means of a thermocouple. The furnace was shielded from air currents, but sufficient opening was left in the shields to provide an adequate air supply for the combustion of the coal. This apparently crude method is open to many objections, but it is surprising how closely the results can be made to check. Out of thirty runs by this method on the same coal, twenty-four agreed within ten degrees.

The accuracy of the method was next tested by experiment on several representative coals. Among the samples tried were bituminous coal, coke, anthracite, and lignite. The values for the glow points of these coals are shown in the following table:

TABLE 1

VALUES OF GLOW POINTS FOR DIFFERENT COALS

Kind of Coal	District	Glow Point	
		Deg. C.	Deg. F.
Bituminous	Georgetown, Ill.	456	853
Bituminous	Bennett Station, Ill.	496	925
Bituminous	Herrin, Ill.	440	824
Lignite	Manitoba	526	979
Semi-bituminous	Castle Gate, Utah	528	982
Anthracite	Pennsylvania	600	1112
Coke		606	1123

2. *Influence of Various Factors on Ignition.*—Finally an attempt was made to determine the influence of various factors such as ash, moisture, sulphur, size of particles, air supply, etc., upon the temperature of ignition. The results of this work may be briefly summarized as follows:

Ash

The amount of ash present in the coal has no apparent influence on the temperature of the glow point. The ash influences the distribution of heat throughout a mass of coal and may change the time required to bring the coal up to a definite temperature, but, the proper temperature once reached, the coal will glow no matter what the ash content may be.

Moisture

Moisture has a great deal to do with the rate at which the coal is heated up to the point where it will burn, consequently with the shape of the heating curve; but, as the moisture is driven off before the glow point is approached, the actual temperature of the glow point can in nowise be affected by the quantity of moisture present.

Size of Particles

The glow point for all sizes of coal is the same, but in the process of determining this fact some difficulty was experienced in observing the glow points on the fine sizes. The air passages through finer mesh coal are so confined that the interior of such coal is not reached by the air. Furthermore, as its temperature rises, a pile of fine coal begins to cake on the surface, forming a crust that interferes still more with the passage of air to the interior. This made it rather difficult to determine the glow point of fine coal, but the results obtained indicate that there is no difference between its glow point and that of coarser sizes.

Quantity of Air

Glowing of coal is a form of combustion which requires oxygen, but the actual temperature of the glow point is independent of the quantity of oxygen supplied. Obviously, if the coal were heated in an atmosphere devoid of oxygen it would not glow at all, but would undergo destructive distillation. On the other hand, if an atmosphere of pure oxygen were supplied, the rapidity of the combustion would be increased but the temperature of the glow point itself would not be altered.

Sulphur

Separate runs were made on pure pyrite in the effort to determine its ignition temperature, which was found to depend much on its rate of heating. If heated very slowly practically all the sulphur can be driven off in the form of SO_2 without producing any visible flame. If the rate of heating is increased, a blue flame appears at a comparatively low temperature. Unquestionably, therefore, the combustion of sulphur adds to the mass of coal a heat which slightly increases the tendency of this coal to fire, but, as the heat of combustion of sulphur and the percentage of pyritic sulphur present are comparatively low, the total effect of sulphur on the tendency of coal to fire is relatively small.* The experiments appeared to indicate that the presence of pyrite does not alter the glow point of the coal substance itself, but that its only effect is to add its heat of combustion to that of the coal.

Rate of Heating

Coal has a comparatively low heat conductivity, so that, in order to insure a uniform distribution of the heat throughout the sample, external heat must be applied slowly. If the sample is heated too rapidly it will burst into flame before the glow point is reached, due to the ignition at the edge of the pile of small pieces of coal which are at a higher temperature than the average of the pile itself. The best results in the present experiments were obtained when the sample was heated up to the glow point in about twenty minutes' time.

Volatile Products

The ignition points of the various gases volatilized from coal during distillation are fairly well known. In all cases the ignition points of these gases are higher than the glow point of the coal, so that no flame should be observed before the glow point is reached. However, if even a very small particle of coal is heated above the ignition temperature of any of the gases, a flame will result. Thus, in order to successfully determine the temperature of the glow point, it is first necessary to drive off most of the volatile products.

Residual Carbon

It is a well-known fact that various forms of carbon differ in the ease with which they may be ignited. Beyond a doubt the character of the carbon or coke resulting from the partial distillation of the coal has more to do with the temperature of the glow point than any other single factor. The exact reason for this was not determined, nor was any attempt made to differentiate the forms of carbon or coke resulting from various coals tested. Furthermore, when a coal starts to glow it is not yet completely coked. Samples of coal quenched at the glow point show that the individual particles have lost their angular shape and are only partly cemented together.

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* "The Spontaneous Combustion of Coal." Univ. of Ill. Eng. Exp. Sta., Bul. 46, 1910

3. *Acknowledgments.*—The author wishes to express his appreciation of the manner in which Mr. Frank Stamberg carried on the work of the investigation in the laboratory. Several of the samples of coal were furnished by Professor S. W. Parr of the Department of Chemistry of the University of Illinois, who also made valuable suggestions both regarding method of procedure and interpretation of results. Professor H. H. Stoek of the Department of Mining Engineering of the University of Illinois also furnished many helpful suggestions, and it was through his encouragement and assistance that the work progressed.

The work was done in the laboratory of the Department of Mining Engineering of the University of Illinois.

II. IGNITION TEMPERATURE

4. *General Significance.*—Under ordinary conditions the ignition temperature of a fuel is less important than the maintenance of combustion. No matter how large the mass of fuel may be, whether it is in a blast furnace, under a boiler, or in an ordinary household furnace, the origin of the heat necessary to raise this fuel to the ignition temperature is in the kindling which starts the fire. The flame of the match which ignites the kindling is at a much higher temperature than the ignition point of the coal; hence, the starting of an ordinary fire is a process depending upon the application of an outside source of heat, always at high temperature, to some portion of the mass of coal until a part of the coal itself burns and propagates heat to the remainder. At some time during this process the outside source of heat may be removed as, for example, when the kindling is completely burned. At all times during this process there is a zone of heat production at a temperature well above the ignition point. Whether this source of outside heat is in the form of kindling or in a form similar to the heat in the walls of a coke oven, it is only necessary to supply the heat until a certain part of the coal is burning and raising the temperature of the rest of the mass by its own combustion.

It is possible that the ignition temperatutre of fuel might assume a greater significance than this in connection with problems involving burning of pulverized coal. For example in the cement industry, where pulverized coal is used largely for heating the kilns, trouble is sometimes experienced when the supply of coal is temporarily cut off and the kiln allowed to cool. In resuming the supplying of coal to the cooled kiln, an explosive atmosphere, apt to cause damage, is sometimes produced. It is suggested that this may be prevented by supplying temporarily a fuel with a lower ignition temperature.

It is, however, in connection with the storing of coal in piles that the ignition temperature takes on a real significance. According to Professor S. W. Parr* and others, coal is actually burning at all times and even at low temperatures. If the heat thus produced is sufficiently confined and air is supplied to the coal at the proper rate, the temperature of the interior of the mass will rise. As the temperature rises the rate of combustion also rises, and, after a certain safe temperature is passed, the process goes on quite rapidly until burning or flame results.

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* "The Spontaneous Combustion of Coal." Univ. of Ill. Eng. Exp. Sta., Bul. 46, p. 50, 1910.

One of the original objects of this investigation was to determine the influence of the temperature of ignition of the various coals upon their liability to fire while in storage. At first thought it would seem that the ignition point would have a very important bearing on the tendency to fire. As a matter of fact, however, the glow points determined, with a few notable exceptions, bear no relation to the liability to fire. The same is true if some point on the heating curve other than the glow point is assumed to be the temperature of ignition. For example, the point at which the coal burns with a blue flame is a rather definite point which shows some variation with different coals; but even this variation bears no relation to the liability of the coal to fire.

However, these results are not in any sense conclusive, since several factors known to affect the firing qualities were not considered. For example, Wheeler has found* that the relative ignition temperature of coal depends upon the total oxygen content. Furthermore, experience shows that after coal has been stored for a certain time it becomes more or less immune from firing. Neither the oxygen content of the coal nor the time elapsed since it was mined was considered in the present experiments.

The temperature at which coal bursts into a yellow flame depends so much upon the rate of heating and other conditions, and varies so widely, that it is impossible to establish any quantitative relationship between it and the firing qualities of the coal.

The rate of heating or the shape of the heating curve of various coals differs widely, and this rate is far more important than any temperature which may be chosen as the ignition temperature. The shape of this heating curve is influenced by the volatile matter present, percentage of ash, moisture, sulphur, and oxygen, the air supply, the form of carbon in the partially coked residue, the size of the sample, and, perhaps, by other factors.

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* See Table 7.

5. *Discussion of Ignition Temperature.* In order to arrive at any satisfactory conclusion regarding the ignition point, some idea must first be obtained as to how "ignition temperature" may be defined.

At first thought it would seem that ignition point is the temperature at which self-heating begins and is maintained, but this, in the case of coal, seems to be a very indefinite point.

In the words of Wheeler:*

> "If one adopts the generally accepted definition of Ignition Temperature as applied to gaseous mixtures, (that is, the temperature at which self-heating of the mixture begins to take place) the ignition temperature of most coals is the normal atmospheric temperature.
>
> "A distinction between one coal and another could, no doubt, be made by determining the time taken for this self-heating from atmospheric temperature to result in flame under standard conditions.
>
> "It may also be of value to know at what temperature a given coal must be heated in air, under specific conditions, in order that it shall burst into flame, and this temperature may, perhaps, be regarded as the 'ignition temperature,' the preliminary self-heating (corresponding with the 'fire-flame period' with gases) being ignored. Experimental difficulties, however, stand in the way, both of determining accurately the moment when flame appears in a mass of coal, and of ensuring that in a comparative test with different coals all the factors, on which the appearance of flame depends, are maintained constant.
>
> "From a practical, as well as a theoretical point of view, it is sufficient to know at what temperature, under standard conditions, different coals begin to react with oxygen so rapidly that the ultimate appearance of flame is assured."

Again J. F. Cosgrove† states that the ignition temperature of coal "is, of course, the temperature of ignition for the fixed carbon of the coal, because the gases are roasted out of the coal before the coke burns."

J. S. S. Brame in his book on "Fuels" discusses the matter as follows:

> "For active combustion to be initiated, a definite temperature, at least, must be maintained. The ignition point of all substances occurs no doubt at a

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* "The Oxidation and Ignition of Coal." Jour. Chem. Soc., Vol. 113, p. 949, 1918.

† "Coal, Its Economical and Smokeless Combustion."

fixed temperature, but many conditions influence the ease with which the combustion may be started, mass and fineness of division being the most important. While a given coal, in a finely divided condition, may ignite at a low temperature, a lump of the same coal will require considerable heating before it will take fire, due to the smaller surface exposed to the air in proportion to the mass, which carries away the heat.''

F. H. Sinnatt and Burrows Moore* give the following definition: ''Temperature of ignition in oxygen is defined as that temperature to which fuel must be raised for its ignition to take place without the aid of external agents of inflammation.'' In this definition, as in some of the others, the temperature to which the coal is heated is used as the ignition point, but no account is taken of the heat due to the coal itself burning. In the experiments carried on by Sinnatt and Moore the time required to heat the samples of coal to glowing, flaming, and exploding temperatures was recorded, but in all cases these temperatures applied to the external source of heat and not to the temperature of the coal itself. As a matter of fact, if the coal is heated to a certain temperature it will carry on oxidation and combustion at an increased rate without any visible sign of such combustion, such as glowing or flaming. The temperature to which coal must be raised in order that it may carry on its own increase in temperature up to the ignition point is generally considered to be the ignition temperature.

6. *Various Suggested Ignition Temperatures.*—In the small amount of experimental work and printed discussion on ignition temperature of coal, several distinct points have been mentioned as possible criteria for comparing various coals.

(1) The temperature at which self-heating begins.

(2) The temperature to which coal must be raised in order that it may unite with oxygen and burn.

(3) The temperature (zone) at which rapid self-heating begins.

(4) The temperature to which coal must be raised in order that it may maintain its own combustion.

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* Jour. Soc. Chem. Ind., Vol. 39, pp. 72-78, March 31, 1920.

(5) Definitions on a time basis, such as the time required for a given external temperature to ignite the coal or cause it to glow.

(6) The temperature of the glow point.

(7) The blue flame temperature.

(8) The crossing point of the outside heat, coal-heat curves. (Wheeler.)

The following objections may be raised to all of these methods of defining the ignition temperature.

(1) The temperature at which self-heating begins is actually atmospheric temperature, since coal is combining with oxygen even at low temperatures and producing small amounts of heat.

(2) A definition referring to the temperature at which coal will unite with oxygen and burn in reality defines the flaming temperature, as flaming is the usual visible evidence of combustion. The flaming of a coal, however, depends upon the character of the volatile matter driven off, which varies with the conditions of the heating. The flaming temperature, therefore, is very uncertain.

(3) The heating curves of all coals show an abrupt rise in the neighborhood of 200 deg. C; but there is no definite temperature during this change which may be assumed as the point where rapid self-heating begins. Hence this zone must be noted by its upper and lower limits or by its average temperature, either one of which would be rather indefinite.

(4) The temperature to which coal must be raised in order that it may sustain its own combustion without outside heat is one of the most logical points which could be chosen as an ignition temperature. However, as has already been pointed out, this temperature depends much on the size of the particles and on the air supply. In addition to the necessity of providing different values for different sizes of coal, there would probably be grave experimental difficulties in measuring the exact temperature at which combustion of the coal would be self supporting.

(5) Any definition based on time would not be a true definition but would simply serve as a criterion for comparing various coals.

(6) The glow points were found to be much easier to check with a fair degree of precision than was at first expected. As they are independent of variables such as heating rate they may be measured in various kinds of furnaces with the assurance that the results obtained will check any other operator's figures on the same coal. On the other hand the principal objection to the glow point as a definition lies in the fact that, before it is reached, a large portion of the volatile combustible matter has been driven off: and some of this matter driven off at the very lowest temperatures may have an ignition temperature lower than the glow point. In the experience of all investigators on the subject of ignition point, however, the semi-coked residue has been found to have a lower ignition temperature than any of the evolved gases. The temperatures of ignition of such gases as CH_4, CO, etc., which are known to be given off by coal at low temperatures, are all higher than the glow points observed. True, it was found that the glow point bore practically no relation to the liability to fire while in storage, but this corresponds with the experience of Winmil and Graham,* who state that the most inflammable coal is not necessarily the most liable to spontaneous combustion.

(7) If after all the volatile matter has been driven off from a coal, the residue is brought up to a sufficiently high temperature it will flame with a non-luminous flame. Athough this matter has not been investigated thoroughly it is probable that coals differ very slightly in the temperature at which this blue flame will appear. This is asserted in view of the fact that, in order to obtain the blue flame, practically all the volatile matter must be driven off previously.

(8) Wheeler has shown that the crossing point of the outside heat curve and the coal heating curve as a relative ignition temperature can be easily checked in his type of apparatus.† The objection to any methods of this kind, however, is that the results cannot be duplicated except in identical apparatus and perhaps by the same observer. Furthermore this "relative temperature" marks no definite event in the heating process. It occurs some-

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* I. & C. Tr. Rev., Vol. 89, p. 336.

† See page 35 for a description of Wheeler's work.

time after the coal has begun to heat rapidly enough to cause the curve to cross the furnace curve but it does not mark the beginning of this rapid rise.

7. *Explanation.*—It is highly probable that the ultimate explanation of ease of firing, temperature of the glow point, or ignition temperature, however it is defined, must be sought in the chemical composition of the volatile products and the semi-coked residue. Brame in "Fuels" states: "Generally the proportion of hydrogen present is the determining factor in ignition point of fuel, this being well illustrated in the case of charcoal, which, if carbonized below a visible red heat, ignites at 427 deg. C., while if carbonized at a bright red heat, 900 deg. C., it ignites at approximately 537 deg. C."

In Wheeler's opinion adsorbed oxygen is the determining factor. Having made various tests on coal of different total oxygen percentages, he states that the relative ignition temperatures decrease as the oxygen content increases, but that no relationship exists between ignition temperature and other analytical data. Wheeler also makes the statement that the oxygen content given in his figures must be regarded as that of the saturated coals, because the treatment to which they had been subjected involved exposure. He also shows that a coal which takes up a large amount of oxygen shows self-heat at a lower temperature than low oxygen coals. He says, "The term self-heat may be seen to have its true significance so far as the highly oxygenated saturated coal is concerned if one considers that the loosely combined oxygen may, at a higher temperature, desire a more permanent attachment. If a reaction of the kind does take place. evolution of heat should be observed on a saturated sample if heated in an atmosphere of nitrogen."

Professor Parr suggests, in personal communication, that it is probable that oxygen adsorbed after the coal is mined has an important bearing on the glow point and the firing qualities of the coal. In his opinion, also, the nature of the gases given off just previous to the glow point will have an important bearing on the temperature of the glow, and these gases will probably show some alteration after the coal has been stored for some time. He therefore suggests that one series of experiments be undertaken to determine the chemical composition of such gases and the effect of storage upon the glow point; and another to compare the glow points of freshly mined samples with those of samples saturated with oxygen.

Hollings and Cobb* obtained heating curves which show a marked exothermic reaction in nitrogen between 150 deg. and 250 deg. It is significant that the range 150 to 250, over which exothermic reactions in an inert atmosphere are manifest, should correspond so closely with the range over which rapid self-heating occurs in a stream of air.

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* Jour. Chem. Soc., Vol. 107, p. 1109, 1915.

III. Description of Methods Used by Others

8. *Method of Sinnatt and Moore.*—Fig. 1 shows the apparatus used by F. S. Sinnatt and Burrows Moore for the determination of the relative temperatures of spontaneous ignition of solid fuels.

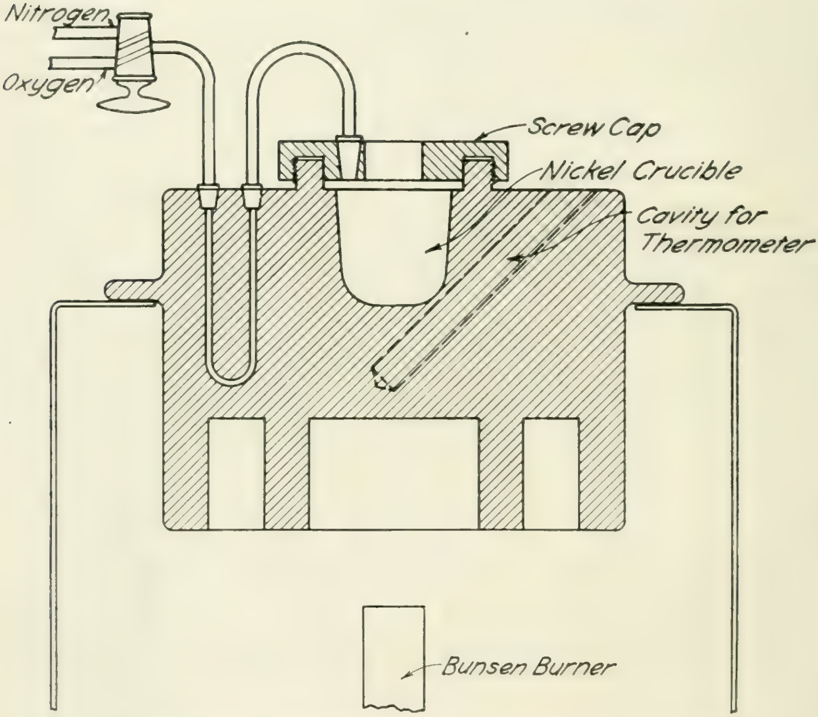

Fig. 1. Apparatus Used by Sinnatt and Moore for Determining Ignition Temperature

The apparatus consists of a heavy cast iron vessel in which an enclosed nickel crucible may be raised to any desired temperature. The whole apparatus is enclosed in an asbestos-lined box to eliminate the influence of drafts. Before entering the combustion chamber the oxygen or other gas traverses the U-shaped tube bored into the casting.

The length of this pre-heating tube is sufficient to raise the temperature of the oxygen to that of the casting before it comes into the combustion chamber. A thermometer is placed in a cylindrical chamber arranged so that the mercury bulb is immediately below the combustion chamber. The casting is deeply grooved at the bottom in order to increase the surface exposed to the flame of a large bunsen burner which is used as a source of heat. The oxygen is dried by passing it through concentrated sulphuric acid. In certain experiments nitrogen was used to sweep out the crucible and a three-way tap was attached to the ignition meter so that nitrogen or oxygen can be passed into the combustion chamber as required.

Pulverized fuel was used in all these experiments and in each case a measured amount was dropped into the nickel crucible by means of a dropper made of glass tubing.

After the introduction of the sample a time record was kept of the following phenomena: first, commencement of visible active combustion, glowing of the mass; second, ignition as manifested by an explosion or production of flame; and third, extinction of the visible combustion. The special points noted were the time interval before visible combustion or glow occurred and the time interval before explosive or inflammatory ignition occurred. The progress of the combustion was watched through a central hole in the lid and the residue after visible combustion had ceased was examined preparatory to the introduction of a fresh charge. Tests were made on the same coal under various temperature conditions in the apparatus, and Tables 2 and 3 show some representative results obtained by these experiments.

The investigators, Sinnatt and Moore, make the following remarks regarding some of the coals:

> "In the coals examined, with the exception of the two which are known to be liable to gob fires, there appear to be zones of temperature in which spontaneous ignition occurs after a much longer interval. It would appear that the coals yield volatile compounds at these ranges of temperature, which compounds have a higher temperature of ignition. It is suggested that such coals may possess the property at these temperatures of tending to extinguish any heating which may be taking place especially where the coal is in masses and the oxygen only in contact with the surface. Where, however, coal does not evolve such compounds, no such self-damping action is possible, and the result of the oxidation

TABLE 2

COMBUSTION PHENOMENA OBSERVED WITH COAL LIABLE TO PRODUCE GOB FIRES

Time in Seconds for			Temp. Deg. C.	Remarks
Extinction	Glow	Ignition		
52	29	35	228	
55	20	35	230	
45	13	15	243	Very powerful explosion.
44	13	14	250	Powerful explosion, yellow flame.
40	11	14	267	Very powerful explosion, yellow flame.
40	10	14	278	
38	10	14	290	
39	10	14	300	Very powerful explosion, yellow-white flame.
38	9	13	310	
38	8	14	320	
25	6	9	330	
37	4	10	340	
35	3	3-14	350	Very powerful explosion at 14.
36	5	10	362	Very powerful explosion, yellow-blue-white flame.
30	2	2	378	
40	1	1	386	

TABLE 3

COMBUSTION PHENOMENA OBSERVED WITH ANTHRACITE

Time in Seconds for			Temp. Deg. C.	Remarks
Extinction	Glow	Ignition		
. .	. .	. .	246	
. .	. .	. .	248	
113	45	. .	250	
125	32	32	256	
140	31	31	258	Yellow flame.
136	29	29	267	
120	23	23	278	
130	17	17	287	
90	19	. .	296	
105	17	20	300	
110	15	45–50	310	
100	12	41	321	Yellow-white flame
110	12	14	332	
108	9	12	342	
90	6	6	350	
100	6	28–35	362	
102	6	. .	368	
120	7	50–58–70	384	Yellow-white flame at 50; blue flame at 70.
83	4	38	388	
106	5	52	390	Yellow-white-blue flame.
121	4	35	394	
111	6	17	400	Very powerful explosion.
90	5	5	412	

would be an additive one, consisting of the glowing of the solid substance and the ignition of the volatile constituents.

"In order to obtain an idea of the minimum temperature at which substances would either glow or ignite a time limit of four minutes was adopted, and the lowest temperature at which substances would either glow or ignite within this period was taken as the temperature of glowing or ignition. The time decided upon was sufficiently long for most practical purposes."

The values obtained for a number of substances are given below.

TABLE 4

MINIMUM TEMPERATURES FOR IGNITION IN OXYGEN

Substance	Minimum Temperatures in Oxygen, Deg. C.			
	Glowing	Ignition	Holm's Figures	Brame's Figures
Anthracite coal	250	258	440	500
Hoo cannel coal	225	230	....	....
Wigan Arley coal	184	188	....	370
Wigan yard coal	219	219	....	to
Indian coal	242	242	....	425
Coal liable to gob fires, No. 1	228	228	....	....
Coal liable to gob fires, No. 2	228	228	....	....
Gas coke	Above 398	Above 398		
Coalite	Above 396	Above 396	....	....
Wood charcoal	248	248	....	....
Irish peat	300	300	....	....
Cellulose	324	324	360	....
Fusain	367	367	....	....
Paraffin wax	....	246	310	....

The investigators continue:

"Early in the experiments it was found that the degree of fineness of the particles of the fuel had a considerable influence upon the minimum temperature at which spontaneous glowing would occur, and preliminary observations have been made to ascertain broadly the influence of the fineness of the fuels examined. The results are recognized as being quite preliminary but, as they show that the degree of fineness is a most potent factor, it is desirable to place them on record."

An attempt was likewise made to ascertain the influence of the volatile matter present in the fuels on the temperature at which spontaneous ignition occurs. In order to do this the apparatus was

TABLE 5

MINIMUM TEMPERATURE AT WHICH SPONTANEOUS GLOWING OCCURS

Description	Degree of Fineness		
	1/200	1/100 to 1/200	Mixture of 2 parts 1/200 and 1 part 1/100-1/200
	Temp. Deg. C.	Temp. Deg. C.	Temp. Deg. C.
Indian coal........	240	Above 380	242
Hoo cannel.........	229	Above 390	277
Yard coal..........	219	Above 396	233
Coal liable to gob fires, No. 1	232	348	...
Coal liable to gob fires, No. 2	225	298	...

arranged so that nitrogen could be passed into the combustion crucible through the three-way tap shown in the diagram, until the volatile matter had been driven off from the coal. The procedure was as follows: The ignition meter was raised to any desired temperature, the nitrogen turned on, and a charge of coal introduced from the dropper. The coal was kept for 15 minutes in the atmosphere of nitrogen. At the end of this period the supply of nitrogen was interrupted and oxygen introduced into the crucible. The time interval before glowing or ignition occurred was then determined. The crucible was at once emptied of any residue and a fresh charge or blank was dropped in, having the oxygen continually flowing. The process was repeated for different temperatures and the results are given in Table 6.

Sinnatt and Moore say further:

> "It may be observed that at temperatures only slightly above the minimum temperatures previously recorded for the occurrence of glowing, all the fuels, except the Hoo cannel, which had been previously exposed to nitrogen failed subsequently to glow or ignite in oxygen within the time interval of four minutes whereas the control samples examined immediately afterwards behaved exactly as previously recorded. The conclusion to be deduced from these observations is that the volatile matter in the coal has considerable influence upon the liability of the fuel to glow. The non-appearance of the glowing in the nitrogen-treated charges appears especially important where the material has only been

heated to temperatures not appreciably above the minimum temperatures previously recorded for glowing to occur. It would therefore appear that the oxidation of the volatile constituents evolved contributes very largely to the heating effect, which leads to the mass of coal becoming heated, and consequently to the production of a glow."

TABLE 6

TIME INTERVAL BEFORE GLOWING OCCURS IN NITROGEN AND OXYGEN

Exposed to Nitrogen for 15 Min. before Admission of Oxygen				No Previous Exposure to Nitrogen before Admission of Oxygen	
Air Dried Sample	No. of Seconds before Glowing	Remarks F—flame E—explosion	Temp. Deg. C.	No. of Seconds before Glowing	Remarks
Arley coal 200 mesh	20	Feeble E at 20 sec. Yellow F.	398	1	More powerful E at 43 sec. Blue F.
	10	Feeble E at 10 sec.	378	1	
	35	Residue partly ash	285	3	More powerful E at 25 sec.
		No glowing	240	13	Explosion at 25 sec.
		No glowing	226	25	
		No glowing; unburnt residue	260	8	
Anthracite coal 200 mesh		No glowing	388	7	Yellow F.
		No glowing	362	12	Feeble E.
		No glowing	325	22	
		No glowing	300	30	
Hoo cannel 200 mesh	8	Feeble E at 12 sec.	382	1	Fairly powerful E.
	10	Very feeble E at 12 sec.	363	3	Fairly powerful E.
	11	Very feeble E.	340	3	No ignition
		No glowing	270	16	No ignition
Irish peat 200 mesh	20	No glowing; residue charred	390	1	
		No glowing; residue charred	385	1	
		No glowing; residue charred	350	2	
Cellulose 200 mesh	15	No glowing; residue charred	396	1	
		No glowing; residue charred	369	1	
		Feeble E at 15 sec.	340	30	Powerful E at 30.
Fusain 200 mesh		No glowing	389	6	Yellow F.
		No glowing	370	10	

9. *Remarks on Method of Sinnatt and Moore.*—In the method just described for obtaining the ignition temperature it is obvious that the temperatures obtained are those of the external source of heat and not of the coal itself. The very small sample used, about 0.0052 g., did not in any way affect the temperature of the apparatus

to an extent which could be measured. It is impossible with this type of apparatus to measure the temperature of the coal itself at any stage in the process. The chief value of the method lies in the possibility of determining the relative ease with which coals fire while in storage, but the reasons for the ease of firing are not made known. It is to be noted in Tables 2 and 3 that the glow points are in all cases reached before ignition takes place.

The results obtained by these men, although taken in a manner very different from that of the present investigation correspond in every way with those recorded later in this paper.

Present experiment, however, determined that the glowing temperatures were far above those recorded by Sinnatt and Moore in their table. This explains why these investigators obtained no glow after the volatile matter had been driven off, as shown in Table 6. It has been found by several investigators that the giving off of the volatile products in an atmosphere of oxygen or air is accompanied by exothermic reactions which add their heat to the mass of the coal. Therefore the explanation of the failure to obtain the glow point is that, after the volatile matter had been driven off in an atmosphere of nitrogen, very little heating effect could be obtained from the exothermic reactions, while the semi-coked residue which was later submitted to the action of oxygen was at too low a temperature to glow, and its combustion in oxygen was going on too slowly to raise the temperature of the coal mass to the glow point. Had the oxygen been admitted at a trifle more rapid rate, the glow would undoubtedly have appeared. In Tables 2 to 6 the temperatures indicated are those of the apparatus. Undoubtedly if the temperature of the coal sample had been measured it would have been found to be considerably above that of the apparatus when the coal glowed.

10. *Method of Wheeler.*[*] —The apparatus used by Wheeler is shown in Fig. 2. It consists of a tube holding the sample of coal, and an electrically heated sand bath surrounding the sample tube. Air is drawn through the sample at a constant rate; the temperature of the coal mass itself and the external sand bath being taken simultaneously. Experiments were made on 40-g. samples of 150-mesh and 240-mesh coal. The temperature of the sand bath was slowly raised

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[*] Jour. Chem. Soc., Vol. 113, p. 949, 1918.

at a uniform rate and the temperatures inside and outside the coal taken at intervals.

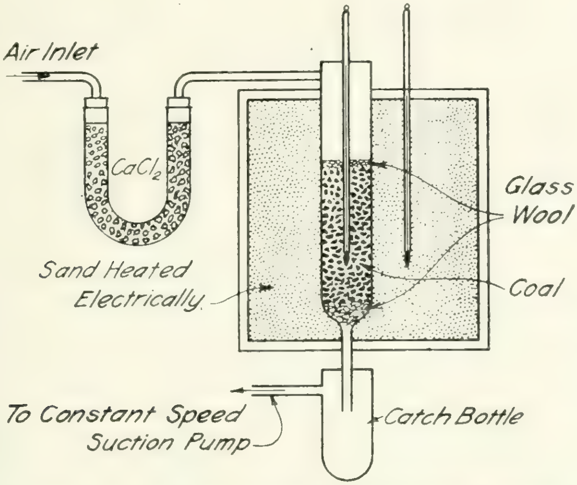

FIG. 2. APPARATUS USED BY WHEELER FOR DETERMINING IGNITION TEMPERATURE

The results of these experiments were plotted as two time-temperature curves, one showing the heating rate of the coal sample and the other showing the heating rate of the external sand bath. In all cases the temperature within the coal eventually began to rise rapidly above that of the sand bath, but, due to the kind of thermometers used, it was impossible to carry the experiment to very high temperatures. It may, however, be assumed that after the heating curve began to rise rapidly it would only be a matter of time before the coal would flame. A curve, characteristic of these results, is shown in Fig. 3.

Wheeler says:

> "It will be seen that the temperature at which the reaction velocity becomes so rapid that inflammation of the coal is imminent is clearly indicated to be at about 205 deg. C. It is evident also from the gradual approach of the two curves that the heat due to the oxidation of the coal was sufficient, despite the loss to outgoing air, to raise the temperature of the coal above that normally due to the heat received from the sand bath when the reaction was proceeding at a temperature of 125 deg. C. The latter temperature might be regarded as

the ignition temperature of the coal, (that is, the temperature at which self-heating begins) under the conditions of the experiment.

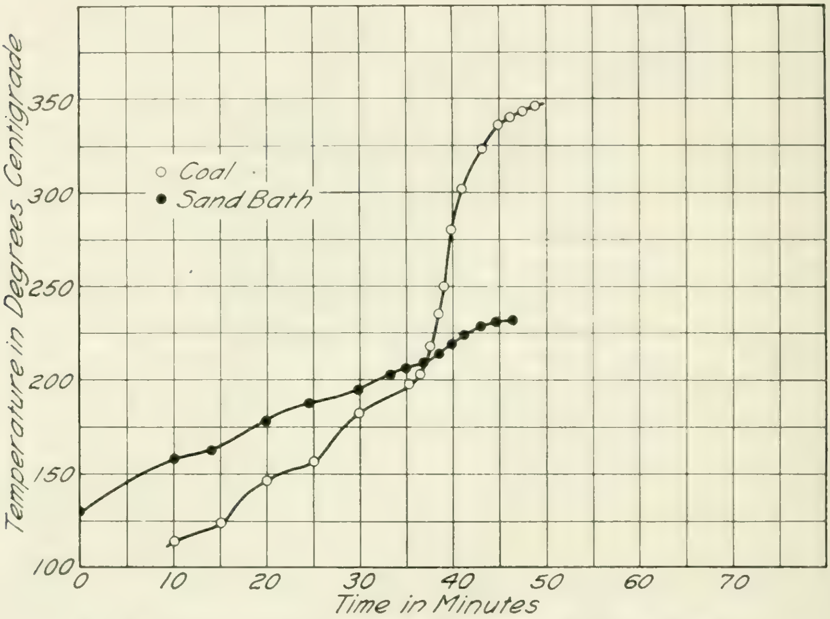

FIG. 3. HEATING CURVES OF COAL AND SAND BATH (WHEELER)

"For comparative purposes, however, it was found preferable to record as the relative ignition temperature the point at which the coal temperature curve and the sand bath curve cross, as this point is clearly defined. It is the temperature at which rapid self-heating begins and is therefore in conformity with Nernst's definition of the ignition temperature for gaseous mixtures."

Wheeler treated several samples of coal having various oxygen contents and determined that the most highly oxygenated fuels are those most liable to self-heat. His results are shown in the following table, in which the oxygen contents were taken on the basis of ash-free, dry coal and must be regarded as those of the "saturated" coals as the treatment to which the coals had been subjected involved exposure.

11. *Remarks on Method of Wheeler.*—The heating curves obtained by Wheeler are similar to those obtained here except for the fact that much higher temperature readings could be obtained by the

use of the thermocouple than with glass thermometer. Wheeler proved unquestionably that the crossing point of the coal-heat, outside-heat curves is a definite point and that he is justified in using this point as a means of comparison for different coals. It is to be noted, however, that these results can be duplicated only in the identical apparatus used by Wheeler, as the shape of the heating curve is affected by the characteristics of the furnace and the conditions under which the experiment is run. Similar to Wheeler's experience, present experiments obtained heating curves showing abnormal tendencies between the range 150-250 deg. C. Fig. 13, page 54, shows this bulge in the heating curve very markedly.

TABLE 7

RELATIVE IGNITION TEMPERATURES FOR VARIOUS OXYGEN CONTENTS

Oxygen Content Per Cent	Rel. Ig. Temp. Deg. C.	Oxygen Content Per Cent	Rel. Ig. Temp Deg. C.
11.1	165	8.3	182
11.1	165	8.1	180
11.1	167	8.0	183
10.6	177	7.6	192
10.5	176	7.4	188
10.3	176	7.3	185
9.9	177	7.0	192
9.9	179	6.7	195
9.9	179	6.6	206
9.5	178	6.4	200
9.2	178	5.6	210
8.8	183	5.4	200
8.8	187	5.1	217
8.7	186	4.9	195
8.6	183	4.7	220
8.5	185	3.9	200

IV. GLOW POINT

12. *Glow Point as Ignition Temperature.*—If a sample of coal is heated carefully so that the rate of heating is uniform throughout the sample it will be found that at a certain very definite temperature the sample will assume a uniform glow. This glow takes place rather suddenly, if one neglects the occasional glowing particles on the edge of the pile, and it is quite certain that it is not entirely due to incandescence of carbon alone, since the color of the glow is much brighter than black bodies assume at this temperature. Furthermore the change from the black coal to the brightly glowing mass is too rapid to be explained by incandescence. After coal has been heated for about twenty minutes under the conditions of the experiment, the change from black to glowing coal takes place sometimes within a half minute of time. With much sulphur present a rather confusing glow occurs earlier than this, but with low sulphur coals the appearance of the glow is quite abrupt. With a few unexplained exceptions the temperatures observed for the glow points of samples of the same coal check fairly closely.

Strange to say there is no very abrupt rise in the heating curve beyond the glow point. To be sure the sample of coal is still rising in temperature, but the rather abrupt rise that might be expected after rapid combustion begins is not apparent. Professor Parr suggests that this might be explained by assuming that at this point adsorbed oxygen, which is probably already partly combined with the coal substance, completely combines with the carbon to form carbon dioxide. The additional heat thus produced is probably not sufficient to alter the temperature at a rate which can be measured by this method. Whatever the explanation of the glow point may be, it occurs at a very definite place in the heating curve and may therefore be accepted definitely as the ignition temperature.

13. *Determination of Nature of Glow Point.*—An attempt to determine the intensity of the action at the glow point was made by submitting a sample of coal above the glow point and one below the glow point to a blast of oxygen. No definite statements can be made regarding the results of this experiment because of the difficulty of

handling a single small particle of coal, determining its temperature, and keeping it from being blown away by the blast. The procedure was to place a small sample of coal in a wire cylinder, raise it above the glow point, and then allow it to cool until the glow just disappeared. As nearly as could be determined the sample was always cooled by the blast of oxygen if its temperature was below that of the glow point, but if it was glowing in but a small part its combustion would rapidly accelerate until it was burning fiercely.

Another experiment made to determine the exact nature of the glow point was to heat a sample of coal in the furnace to about 300 deg., at which time the furnace current was shut off and temperature readings continued. The results of several trials indicated that the coal heated to a temperature just short of the glow point, then the rise stopped. It must not on that account be supposed that further heating is necessary to carry on combustion; indeed, if the concentration of oxygen be increased but slightly the heating will continue even beyond the glow point. The experiment does indicate, however, that the glow point is reached only with a certain amount of difficulty, and that, even if the heating is started, the glow point will not be reached unless a certain concentration of oxygen is exceeded.

These experiments to ascertain the nature and characteristics of glow point furnish in nearly all cases added argument for establishing the glow point as the ignition temperature. The glow point is, moreover, the first visible evidence of combustion, as it occurs in all cases before flame or explosion takes place.

Professor Parr believes that the temperature of the evolved gas has something to do with the glow point and advises a further series of tests to determine the nature of this gas and the influence of weathering on the ignition temperature.

V. DESCRIPTION OF APPARATUS

14. *General Discussion.*—The apparatus used in these experiments is shown in Figs. 4, 5, and 6. It consists of an electric furnace of the resistance type, a motor generator set furnishing direct current, a method of regulating the voltage of this generator by means of shunt field resistance, a Hoskins thermocouple, and a millivoltmeter.

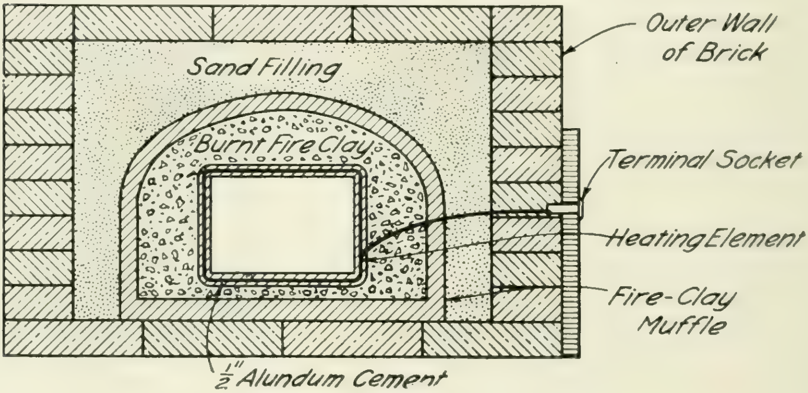

FIG. 4. SECTION OF FURNACE

15. *The Furnace.*—The furnace that was used was made purposely for these experiments and is shown in Figs. 4 and 6. The heating element was made by wrapping 70 feet of special resistance wire, No. 14 B. & S. gauge, having a resistance of 625 ohms per mil-foot and a total resistance of 10.78 ohms, on a reinforced paper form. After the element was wound it was cemented with alundum cement which was allowed to dry before the form was removed. The element was then placed in an ordinary muffle and the space between the wires and the muffle filled in with burned fire-clay. The ends of the wire were brought through holes bored in the muffle to terminals placed on the outside of the furnace. The muffle was then set on a stand and walled in with brick.

A thin layer, ¼ in., of alundum cement was spread on the wires inside the furnace as a lining, and this construction brought the

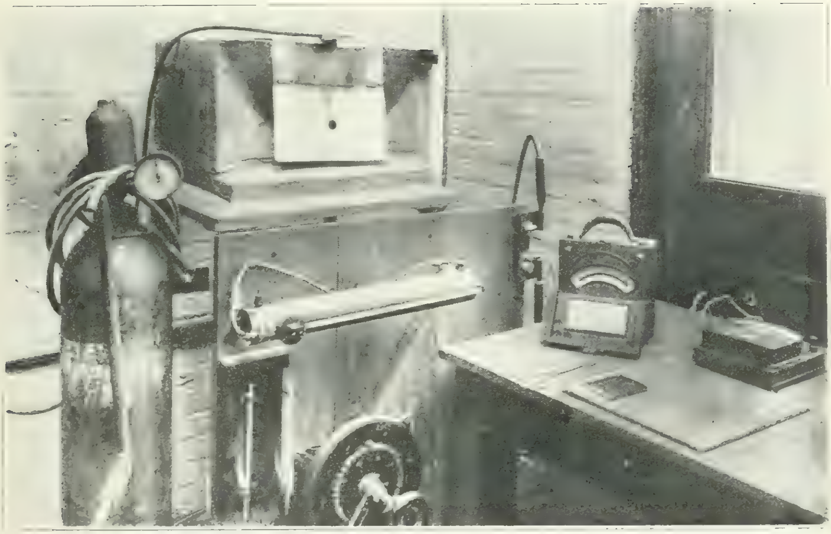

FIG. 6. APPARATUS USED IN PRESENT INVESTIGATION

heating element very close to the interior surface. Because of this the furnace heated rapidly. As the inside was worn off it had to be plastered up occasionally to avoid danger of short circuiting the wires inside the furnace.

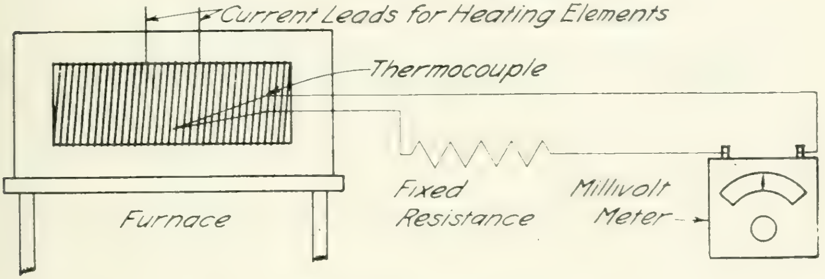

Fig. 5. Wiring Diagram of Furnace

16. *Power Required.*—Because of its ease of measurement and control direct current was chosen as the power for the heating element. The laboratory was already equipped with a 250 volt D. C. compound generator driven by a 440 volt A. C. motor. The voltage of this generator could be regulated by the shunt field control. The switchboard of this set was provided with a field rheostat, but it was found that the successive points on this rheostat gave jumps in the voltage which were larger than desired. A smaller field rheostat, shown in Fig. 6, was therefore wired in series with the switchboard rheostat for the purpose of giving finer adjustments of the voltage. With this auxiliary rheostat it was possible to regulate the voltage so that the current flowing through the heating element was constant at all times.

It was found that 15 amperes gave the best results in the experiments, heating the coal sample as rapidly as possible without showing any great amount of lag due to the low conductivity of the coal. In order to get 15 amperes through the furnace a voltage of about 160 was required, depending somewhat upon the temperature of the heating element. An ammeter was placed in the circuit as shown in the wiring diagram.

VI. TEMPERATURE MEASUREMENT

17. *Method Employed.*—The thermocouple used was a Hoskins base metal couple, the negative side of which is an alloy called Chromel, composed of chromium and nickel, and the positive side Alumel, made of aluminum and nickel. Within the range of temperature desired, this couple has a guaranteed millivoltage of approximately 40 mv. As the millivoltmeter used has a maximum scale reading of 20 mv., it was necessary to use the couple in series with a resistance in order to reduce the scale reading. The couple, resistance, and millivoltmeter were standardized against the melting points of three pure metals.

The placing of a resistance in series with the couple gave the added advantages of high resistance temperature measurements. When the resistance of the couple circuit is extremely low the millivoltmeter reading is affected by the variation of the resistance of the hot wire, imperfect connections, a change in the length of the connections, and many other things. If a high resistance in the circuit is kept at a constant temperature the above mentioned factors will not affect the result appreciably.

The gases produced from the burning of the coal were very corrosive and the couple frequently broke. However, due to the high resistance mentioned above, frequent re-calibrations proved that after the wire was re-welded there was very little difference in the reading for any given temperature.

18. *Calibration of Thermocouple.*—The metals used in the calibration of the couple were aluminum, melting point 659 deg. C., lead, melting point 327 deg. C., and tin, melting point 232 deg. C. These three metals were used because their melting points fell within the limits of temperature most used in this work, and because they gave the most satisfactory calibration points. It was found that zinc gave an uncertain calibration point.

19. *Fixed Junction Correction.*—No attempt was made to keep the fixed junction of the thermocouple at zero degrees, but room temperatures were observed on all the tests and the correction for

room temperature made by the method described by L. H. Adams before the A. I. M. E., September, 1919. In this method the millivolts corresponding to room temperature are added to the observed millivoltmeter reading, giving a value of the voltage which will correspond with the true temperature of the hot end of the couple. In this same paper Adams includes a table showing temperature difference per millivolt with this type of couple. This difference varies slightly for different temperatures, but there is less variation than in noble metal couples or "Constantan" combinations. In other words the calibration curve is nearer a straight line for the Chromel-Alumel couple than for the other types.

TABLE 8

TEMPERATURES FOR VARIOUS MILLIVOLTMETER READINGS

DEGREES CENTIGRADE

M. V.	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14
00	0	46	90	134	179	223	268	312	358	404	451	498	543	587	630
05	2	48	92	136	181	225	270	314	360	406	453	500	545	589	632
10	5	50	94	138	183	227	272	317	363	409	456	503	548	591	634
15	7	53	97	141	186	230	275	319	365	411	458	505	550	594	636
20	9	55	99	143	188	232	277	321	367	414	460	507	552	596	638
25	12	57	101	145	190	234	279	324	369	416	463	510	555	598	641
30	14	59	103	147	192	237	281	326	372	418	465	512	557	600	643
35	16	61	105	149	194	239	283	328	374	421	467	514	559	602	645
40	18	64	108	152	197	241	286	330	376	423	469	516	561	605	647
45	21	66	110	154	199	243	288	333	379	426	472	519	564	607	649
50	23	68	112	156	201	246	290	335	381	428	474	521	566	609	651
55	25	70	114	158	203	248	292	337	383	430	476	523	568	611	653
60	28	72	116	161	205	250	294	340	386	433	479	525	570	613	655
65	30	75	119	163	208	253	297	342	388	435	481	528	572	615	657
70	32	77	121	165	210	255	299	344	390	437	484	530	574	617	659
75	35	79	123	168	212	257	301	347	392	440	486	532	577	620	.....
80	37	81	125	170	214	259	303	349	395	442	488	534	579	622	.....
85	39	83	127	172	216	261	305	351	397	444	491	536	581	624	...
90	41	86	130	174	219	264	308	353	399	446	493	539	583	626	.
95	44	88	132	177	221	266	310	356	402	449	496	541	585	628	....

The table of temperature differences just mentioned could not be employed directly because of the high resistance used, but a similar table was prepared based on Adams's temperature differences and our own calibration, and is given above as Table 8. This table was used in connection with all temperature determinations of the glow points of the various coals. In making the experiments on the heating curve of the furnace and the heating curves of the standard coal a slightly different resistance was used and a separate calibration was made. The table calculated from this is not included. Several recalibrations of the thermocouple during the tests on coals showed practically no deviation from the values given in the above mentioned tables.

Taking all factors into consideration, such as the accuracy of the calibration, loss of heat, lag of the couple, approximations in the calculations, and similar sources of error, the readings are thought to be correct to about 5-8 degrees.

VII. METHOD OF MAKING TESTS

20. *General Procedure.*—The method used in these experiments was as follows: a sample was placed in a flat pan and the thermocouple inserted in the top of the coal so that the junction was entirely submerged, see Fig. 7; the shields were then placed over the furnace and the current turned on; observations of the temperature were made at half-minute intervals and temperatures of glowing, flaming, or other phenomena, were noted.

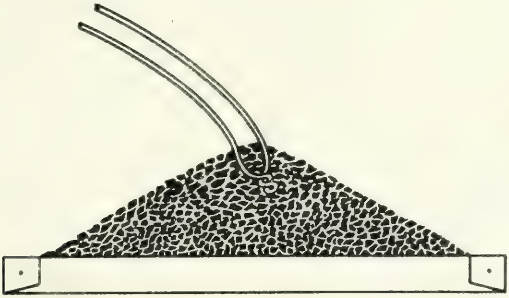

FIG. 7. DEPTH OF IMMERSION OF THERMOCOUPLE

21. *Weight of Sample.*—For the determination of glow point alone it was unnecessary to standardize the weight of the sample as the glow point is independent of the weight. In order to get the heating curves of various coals for purposes of comparison, all the conditions, including the weight of the sample, were standardized. It was found that about 35 g. of coal could be heated at a fairly uniform rate without any great amount of lag in any part of the pile; furthermore this weight was large enough to make possible a good immersion for the couple.

The pan in which the sample was placed was rather flat, with low sides, so that the air had free access to the sample and all parts of the pile could be observed from the front of the furnace.

22. *Setting of Thermocouple.*—It was found rather important to have the proper immersion of the thermocouple in the sample in order to check results. The object of course was to get the

average temperature of the sample itself, or more properly to observe what happened to that coal which immediately surrounded the couple. If the point of the couple was too deep it was in no way certain that what was observed on the surface was also taking place within the pile. On the other hand if merely the tip of the couple was placed in the coal, temperatures observed applied to the gases coming off the coal rather than to the interior of the pile itself.

23. *Distribution of Heat and Temperature Lag.*—In starting these experiments it was uncertain just how much dependence could be placed on temperature measurements made in this manner. The number of degrees by which the thermocouple lagged, the temperature of the coal itself, the distribution of temperature throughout the sample, the lag of the top of the coal pile behind those individual particles next to the hot furnace floor and the difference in temperature between the surface of any particle and its interior were all uncertain factors. In order to eliminate some of the uncertainty regarding the lag of the couple and the distribution of the heat throughout the pile, a cooling curve was taken on a given coal sample whose heating curve had previously been observed, and the glow point on the cooling curve noted. In observing this the temperature at which the coal lost its glow and returned to its black aspect was noted. In every case these cooling curve glow points were the same as those observed on the heating curves. These results indicate that the lag of the couple was very slight and that under the conditions of the experiment the distribution of the heat throughout the sample was fairly even.

The lag of various parts of the coal pile depends on the rate of heating. The first work done was to determine the proper rate of heating and the proper size of particles. It was found that with 35 g. of sample, 15 amperes supplied to the furnace heated the sample at a fairly uniform rate and the glow observed on low sulphur coals was fairly uniform throughout the pile.

No attempt was made to determine the lag of the interior of the individual particles behind the surface because this is relatively unimportant: that is, if we assume that glow point and flame, and as a matter of fact all oxidation, are superficial phenomena, it matters very little what the temperature of the interior of the particle may be. Experiments on large and on very small particles showed that the total amount of surface exposed to the action of oxygen bore no relation to the temperature of the glow point. This fact also gives some weight to the statement that the temperature of the interior of the particle is of no importance.

Some error is probably introduced by radiation of heat from the walls of the furnace to the thermocouple, especially during the early stages of the heating when the furnace is at a higher temperature than the coal. About 8 in. of the couple was exposed in this way, protected only by small, two-holed, porcelain sleeves ⅝ in. long, extending to within an inch of the coal sample. At the glow point, however, the error introduced by radiation is probably small, inasmuch as the walls of the furnace and the sample of coal are both at a temperature higher than the air temperature represented by the heating curve of the furnace.

VIII. Description of Experiments

24. *Heating Curve of Furnace.*—The first work that was done was to compare the heating rate of various currents on the empty furnace. After several quantities of current were tried, both on samples of coal and on the empty furnace, it was concluded that 15 amperes gave the most satisfactory heating curve. To determine the heating curve of the furnace, six runs were made under exactly similar conditions. The furnace was allowed in each case to cool to room temperature and the temperature of the room itself was kept fairly constant by steam heat. The shield on the front of the furnace was in the form of a door hinged at the top, so that, when lowered, it always assumed the same position. This door was made of sheet metal having in the center a $^3/_4$-in. hole through which the action within the furnace could be observed. The shield was not tight

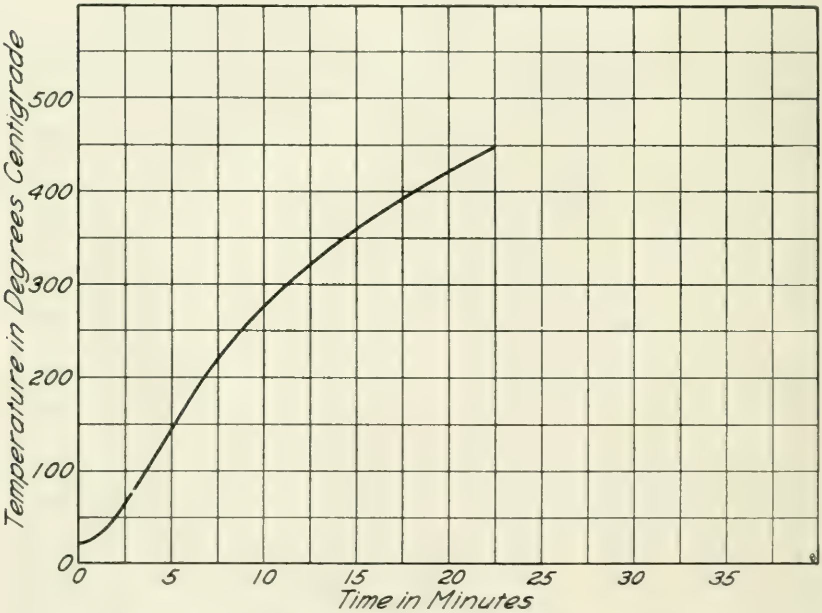

Fig. 8. Average Heating Curve of Empty Furnace

fitting; air could enter the furnace around the edges and through the hole, and also by an opening provided in the back of the furnace for the admission of the couple. The furnace was placed in a corner of the laboratory away from air currents, so that even with openings through which air might pass to cool the furnace, individual observations of heating curves differed but slightly from the average. The maximum deviation of any single heating curve from the average of all was less than 15 deg. The heating curve of the furnace calculated from these six runs is shown in Fig. 8.

25. *Work on Standard Coal.*—A low sulphur and ash coal was hand sorted from available samples in order to provide a relatively pure coal which could be used in standardizing a method for the determination of the glow point. On analysis this coal was found to contain 8.60 per cent ash and 1.25 per cent sulphur. This coal was prepared by crushing to a maximum size ¼ in. and then screening through the following screens: ¼ in. and ⅛ in., and 14, 28, 48, 100, 150, and 200 mesh. Thirty-five gram samples were then heated at a constant rate based on a 15 ampere supply of current to the furnace. Care was exercised to give the same immersion to the couple in every case. Three or four check runs were made on each size to determine the effect of size of particles on the glow point. The results of these runs are shown in the following tables:

TABLE 9

GLOW POINT TEMPERATURES, STANDARD COAL

DEGREES CENTIGRADE

Size of Coal	Run No.				
	1	2	3	4	5
¼ in.—⅛ in.	469	469	463	469	454
⅛ in.—14 mesh	469	473	445	...	...
14—28 mesh	466	467	471	435	...
28—48 mesh	517	510	456	501	...
48—100 mesh	472	470	472	480	...
100—150 mesh	466	468	...	...	...
150—200 mesh	470	466	468	477	...
Through 200	466	472	469	478	...

Average Temperature—472 deg. C.

TABLE 9A

GLOW POINT TEMPERATURES, STANDARD COAL

DEGREES FAHRENHEIT

Size of Coal	Run No.				
	1	2	3	4	5
$\frac{1}{4}$ in.—$\frac{1}{8}$ in	876	876	865	876	849
$\frac{1}{8}$ in.—14 mesh	876	883	833	...	...
14—28 mesh	871	873	880	815	
28—48 mesh	963	950	853	934	
48—100 mesh	882	878	882	896	...
100—150 mesh	871	874	...	...	...
150—200 mesh	878	871	874	891	...
Through 200 mesh	871	882	876	892	...

In at least one run on each of these sizes the furnace was started from the room temperature and observations of the temperature made at half-minute intervals to determine the heating curves. As these heating curves vary but little they were averaged to represent the heating curve of the standard coal. This heating curve was plotted against the average heating curve of the furnace and is shown in Fig. 9.

The results obtained for the glow point on the sizes finer than 100 mesh were uncertain because there soon formed over the pile a crust of glowing coal which made it impossible to determine just what was going on around the thermocouple. This is explained by the fact that in the smaller sizes very much more surface is exposed to the action of oxygen, and the surface of the pile is actually at a higher temperature than the interior. The temperature of this crust, however, is not transmitted to the thermocouple; hence results on the finer sizes are apt to be false. No attention was paid to this initial glow of the crust but as soon as it seemed that the pile was glowing to a depth equal at least to the immersion of the thermocouple the temperature was noted. In every case the results on the fine coal were very uncertain and it would require a great deal of practice before an operator could distinguish the proper point and check himself.

The most consistent results were obtained on the larger sizes from $\frac{1}{4}$ in. to $\frac{1}{8}$ in., and $\frac{1}{8}$ in. to 14 mesh. With this method any one of these sizes could be adopted for use as a standard, and in the subsequent work on individual coals these sizes were used.

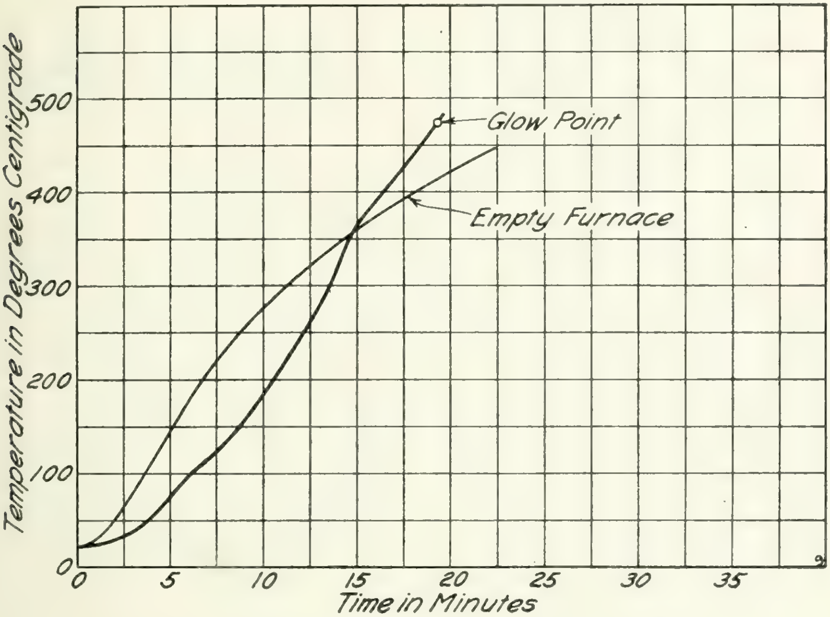

FIG. 9. AVERAGE HEATING CURVE OF STANDARD COAL

26. *Effect of Air Supply.*—To test the influence of various oxygen percentages on the temperature of the glow point the coal was heated under various conditions.

Case I.

The furnace was heated in the ordinary manner but a small stream of oxygen was admitted through the shield. This was done to give the same effect as a larger air supply, without the objectionable cooling effects of a blast of air.

Case II.

The furnace was heated in the usual manner to a temperature of about 300 deg., at which point the current was shut off. Oxygen was then admitted slowly to determine if the glow point could be reached without the action of outside heat.

Case III.

Conditions of Case III were similar to those of Case II except that a blast of oxygen was played on the sample.

In all three cases the action of oxygen was to increase the heating rate. This, however, was done at the expense of accuracy, inasmuch as the pile heated locally and a uniform glow was difficult to obtain.

In Case I a satisfactory glow point was observed by keeping the quantity of oxygen admitted very low. This glow point was found to be the same as when the coal was heated under ordinary conditions.

In Cases II and III when the current was shut off at about 300 deg., it was found that if sufficient oxygen was admitted to the sample the glow could be produced without the assistance of external heat, but if insufficient oxygen was admitted, or no oxygen at all, the coal would heat to some extent but the glow would not appear.

In Case III if the blast of oxygen was strong enough and the coal sample had not yet reached the glow point the latter would actually be cooled off, but if the glow point was first obtained and the coal then submitted to the blast of oxygen it would burn fiercely. This would indicate that the rate at which coal is raised to the temperature of the glow point is largely dependent upon the air supply, but that the actual temperature of the glow point, no matter how it is approached, is not altered.

27. *Effect of Ash.*—To determine the effect of ash on the glow point a sample of coal was mixed with pure shale washed from coal, and run under standard conditions. The following table of results

TABLE 10

EFFECT OF ASH ON GLOW POINT

Glow Points of Coal 35 g. Sample		Glow Points, 25 g. Coal Mixed with 10 g. Shale	
Deg. C.	Deg. F.	Deg. C.	Deg. F.
525	977	526	979
521	970	524	975
512	954	520	968

shows that the temperatures for the glow points under the two conditions were practically the same.

One peculiar feature of this experiment is illustrated in the heating curves shown in Fig. 10. It will be seen that high refuse coal heated more rapidly in the early stages than clean coal, but at a retarded rate towards the end of the process. This retardation was due to the fact that the quantity of coal in this experiment was less by 10 g. than in the other test: hence the quantity of heat derived from the coal after self-heating began was less. The rapid heating of the high refuse material early in the experiment may be explained by all or any of the following considerations:

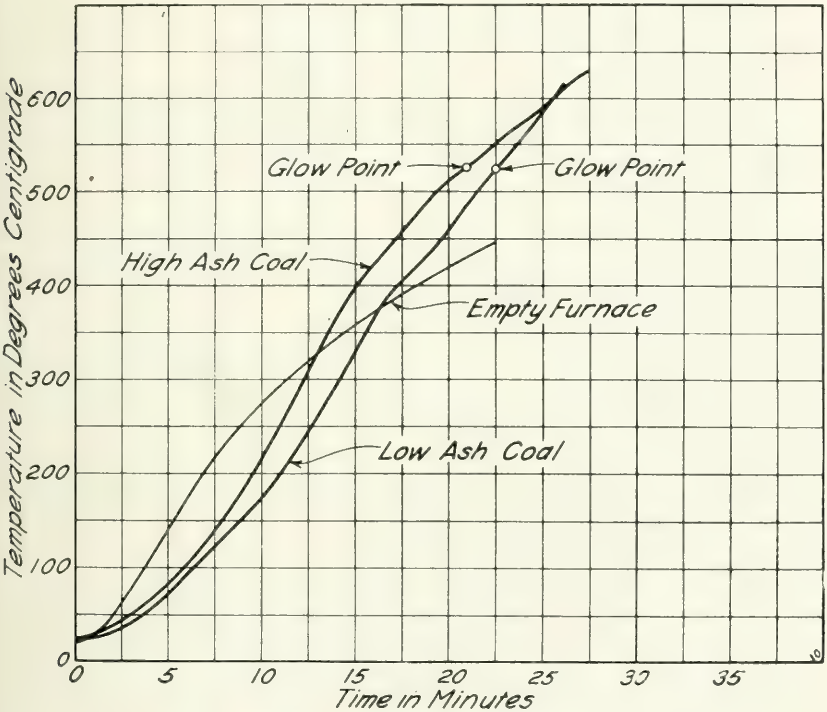

Fig. 10. Effect of High Ash on Heating Curve

(1) Shale has a lower specific heat than coal. No figures on the specific heat of shale are known, but similar substances, such as

brick, kaolin, and ashes, have specific heats ranging from 0.120 to 0.224, whereas that of coal is given by the same authorities as 0.314. To be sure the relative thermal conductivities of these substances enter into the discussion to some extent since it is a rate of heating that is being observed. From the best figures obtainable the conductivity of coal is approximately double that of shale, which fact would tend to offset the opposite relation of the specific heats to some extent.

(2) Pure coal requires more heat in the early stages to drive off moisture and other volatile matter.

(3) In the high refuse samples, more air is supplied to the individual particles or the air which reaches any particular piece is higher in oxygen than in the case when the coal particles are more concentrated.

If the true explanation for this rapid heating of high ash coal could be found it might explain the prevalence of gob fires, refuse pile fires, and fires in piles of dirty coal.

28. *Effect of Sulphur.*—The effect of sulphur was tested in like manner by adding to the coal an amount of pyrite equivalent to about 6 per cent sulphur in the sample, the particles of pyrite being the same size as the particles of coal. As this sample was heated, the individual particles of pyrite and the coal just adjoining them glowed at a comparatively low temperature. Undoubtedly these individual particles were at or above the temperature of the glow point but their accumulated heat had a negligible effect on the temperature of the entire mass. From this it was assumed that the glow point of the coal substance proper is independent of the percentage of sulphur. It must be admitted, however, that this constitutes an objection to the use of the glow point as a criterion for ignition temperatures since it is well known that sulphur in pyrite burns at a lower temperature than coal and consequently assists in its combustion and ignition.

29. *Effect of Moisture.*—The effect of moisture was studied by saturating a 35 g. sample of coal and running it under standard conditions. As in the case of the experiments on the high ash coals the tabulated results show that the glow points for the high moisture samples are practically the same as for the original coal.

TABLE 11

EFFECT OF MOISTURE ON GLOW POINT

Glow Points of Coal		Glow Points of Coal Saturated with Moisture	
Deg. C.	Deg. F.	Deg. C.	Deg. F.
526	979	515	959
525	977	529	984
512	954	526	979

Fig. 11 shows the heating curve under these conditions. In this curve the evaporation of the moisture is clearly indicated at 100 deg., but in spite of this retardation in heating the coal reaches the glow point sooner than that which was not dampened. In this

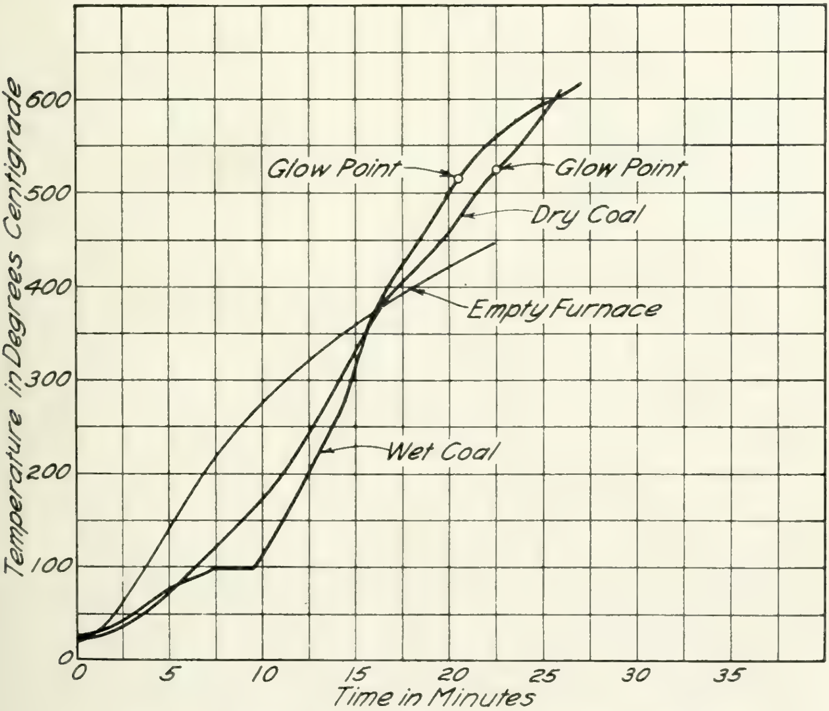

FIG. 11. EFFECT OF HIGH MOISTURE ON HEATING CURVE

experiment the coal showed an unusual tendency to flame before the glow point was reached. Several trials had to be made before the glow point could be obtained prior to the flame. It would seem, therefore, that there is some weight to the contention that better combustion is obtained from coal which has been previously moistened. Certainly in these experiments the moistened coal was more inflammable.

IX. Results and Discussion

30. *Tests on Ten Different Coals.*—In addition to the standard coal ten different coals were tested and their glow points determined as shown in Table 12.

Table 12

Glow Points of Coals Tested

Coal from	Glow Points		Average		Difference in Deg from Average
	Deg. C.	Deg. F.	Deg. C.	Deg. F.	Deg. C.
Herrin, Williamson County	433	811	440	824	—7
	451	844	...	...	11
	437	819	...	...	-3
Georgetown, Vermilion County	453	847	456	852	—3
	453	847	...	...	—3
	466	871	...	...	10
	451	844	...	...	-5
Bennett Station, Vermilion County	498	928	496	925	2
	494	921	...	...	—2
Old Ben No. 8	526	979	521	970	5
	525	977	...	...	4
	512	954	...	...	-9
Lignite, Manitoba, Canada	535	995	526	978	9
	522	972	...	...	—4
	520	968	...	...	—6
Castle Gate, Utah	535	995	528	982	7
	524	975	...	...	—4
	525	977	...	...	-3
Lovington, Moultrie County	526	979	530	986	—4
	534	993	...	...	4
Lincoln, Logan County	559	1038	558	1036	1
	557	1035	...	...	—1
Anthracite, Pennsylvania	598	1108	600	1112	-2
	598	1108	...	...	—2
	605	1121	...	...	5
Coke	607	1125	606	1123	1
	607	1125	...	...	1
	605	1121	...	...	—1

The average glow points shown in Table 12, with the exception of those for anthracite and coke, show no apparent relation between the glow point and the liability to fire while in storage. For example the lignites and the coal from Castle Gate, Utah, which acts very much like a lignite in coking experiments, both have a comparatively high glow point, whereas the coal from Herrin, Williamson

County, Illinois, which ordinarily shows no unusual tendency to fire while in storage, has the lowest glow point of all the coals.

31. *Determination of Various Glow Points.*—So far in these experiments no attempt had been made to determine the oxygen of the coals nor the amount of oxygen absorbed after the coal was mined. It had been attempted simply to develop a method for the determination of ignition temperatures and to show that this method would give consistent results. Wheeler had shown that relative ignition temperature varied with the total oxygen content. Before making any positive statements, therefore, particularly concerning the relation between the glow point and the tendency to fire in storage, it was necessary to make a series of tests determining the glow point of the same coal both freshly mined and after it had been exposed to the action of oxygen for various periods.

Coal No. 1.

(Herrin-Watson Coal Co., Herrin, Williamson County, Illinois.)

This coal had the lowest glow point temperature of any of the coals tested. Three runs were made on the ¼-⅛-in. size, giving results as shown in the table. The heating curve showed no unusual characteristics and is not included.

Coal No. 2.

(Sharon Coal Company, Georgetown, Vermilion County, Illinois.)

In taking the heating curve of this coal it was observed that apparently strong exothermic reactions were taking place before the glow appeared. In order to be sure of this fact and to see how nearly the heating curves could be duplicated, a second curve was taken on this coal. The two heating curves plotted against the furnace heating curve are shown in Fig. 12.

Coal No. 3.

(Bennett Station, Vermilion County, Illinois.)

The peculiarity of this coal may be seen in the heating curve, Fig. 13, which displays rather unusual evidence of exothermic reactions before the glow point is reached. In this particular coal the glow appears on a rather steep portion of the curve, whereas in most of the other samples the glow point marks a flattening of the curve rather than a steeper inclination.

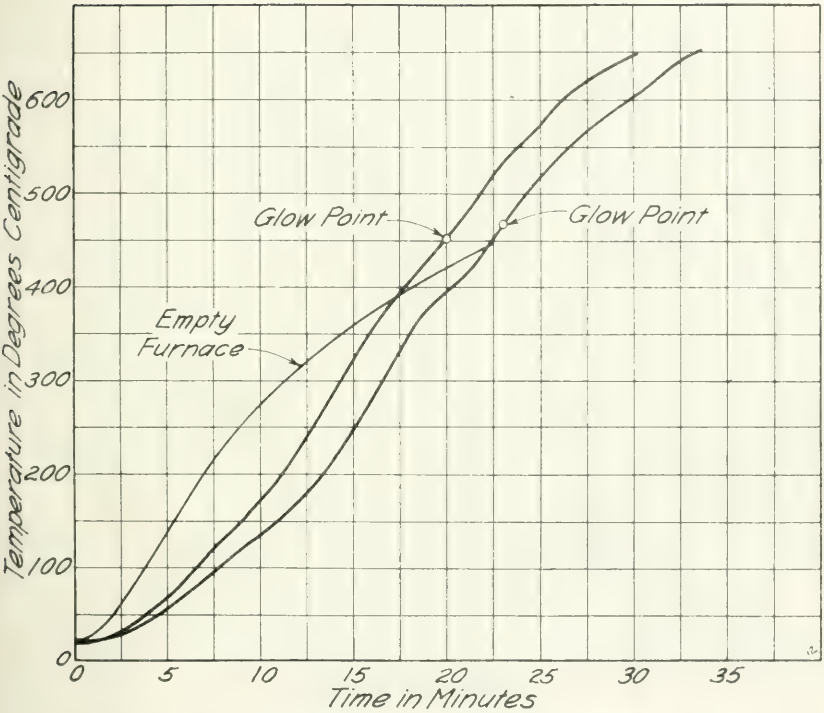

FIG. 12. HEATING CURVES OF COAL NO. 2, SHOWING SIMILARITY OF CURVES FOR CONSECUTIVE RUNS

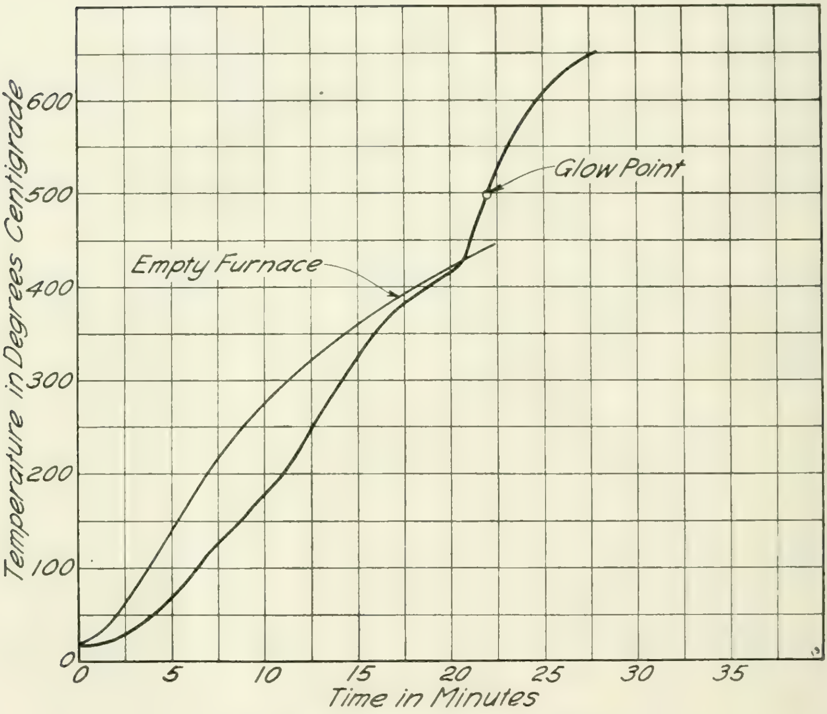

Fig. 13. Heating Curve, Coal No. 3

Coal No. 4.

(Old Ben Corporation, Mine No. 18, Johnson City, Illinois.)

This coal was used in the experiments to show the effect of high ash and high moisture on the heating curve. The heating curve may be seen in Figs. 10 and 11.

Coal No. 5.

(Lignite from Manitoba, Canada. Heating Curve, Fig. 14.)

Up to a temperature of about 150 deg. this curve is comparatively flat, as could be expected from the nature of the coal. During this range moisture and other volatile matter requiring heat for vaporization were being driven off. After this process was complete, however, there was a large discrepancy in temperature between the coal and the furnace; consequently the coal heated rapidly to the glow point.

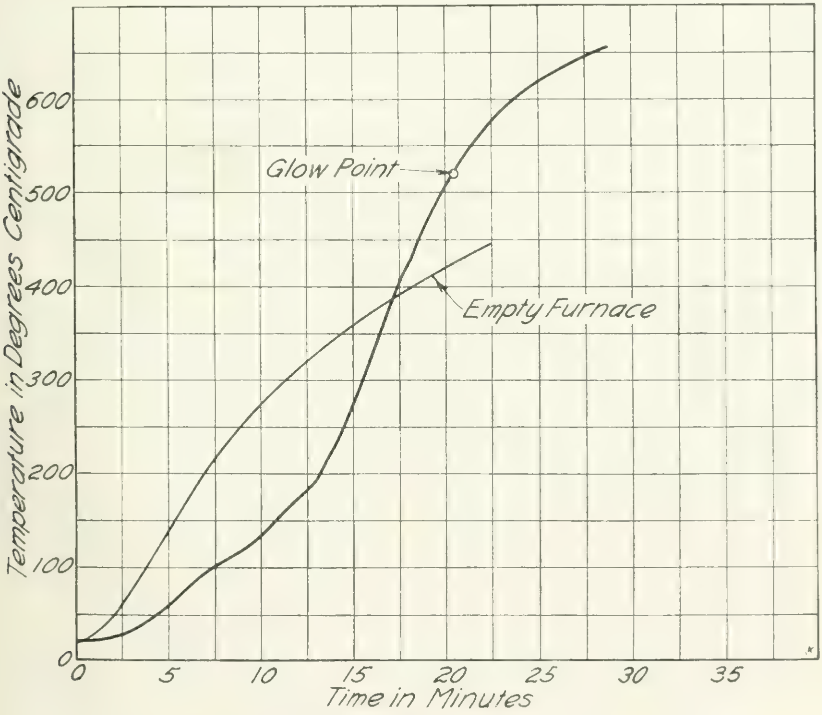

FIG. 14. HEATING CURVE, COAL No. 5

Coal No. 6.

(Bituminous Coal. Castle Gate, Utah. Heating Curve, Fig. 15.)

In coking experiments made under the direction of Professor Parr this coal acted very much as a lignite. The moisture content, however, is only 3.5 per cent, which is much less than that of lignite; consequently the lower range of the curve does not show the flattening tendency that is apparent in the lignite curve. As may be seen in Table 12, the glow point of this coal was practically the same as that of the lignite.

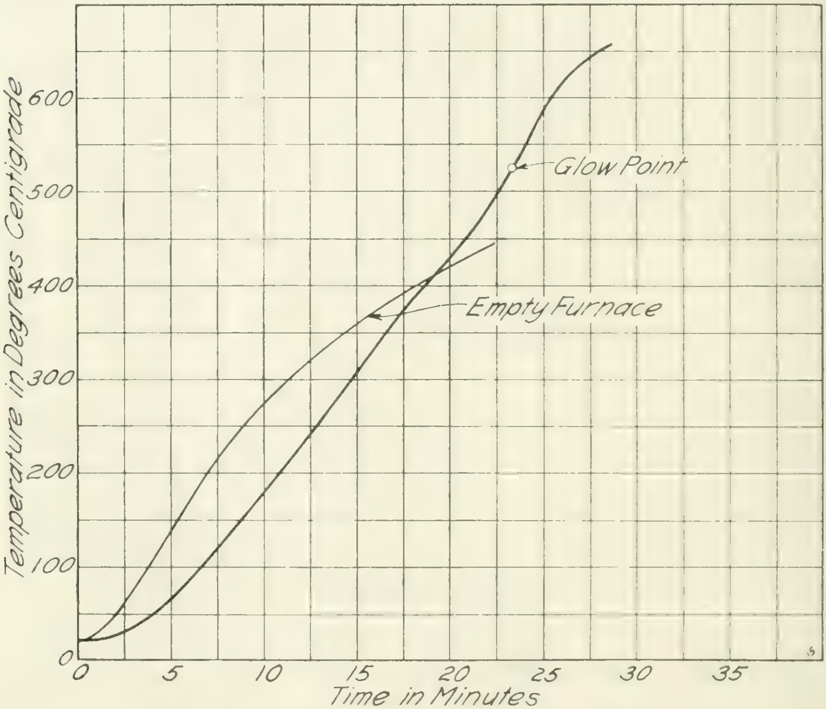

FIG. 15. HEATING CURVE, COAL NO. 6

Coal No. 7.

(Lovington, Moultrie County, Illinois. Heating Curve, Fig. 16.)

Coal No. 8.

(Lincoln, Logan County, Illinois. Heating Curve, Fig. 17.)

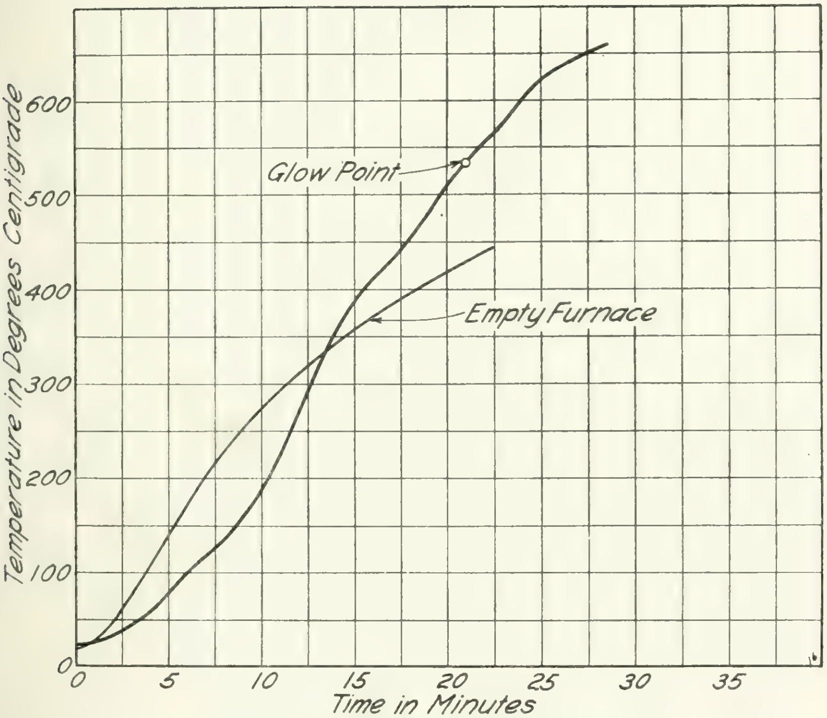

FIG. 16. HEATING CURVE, COAL NO. 7

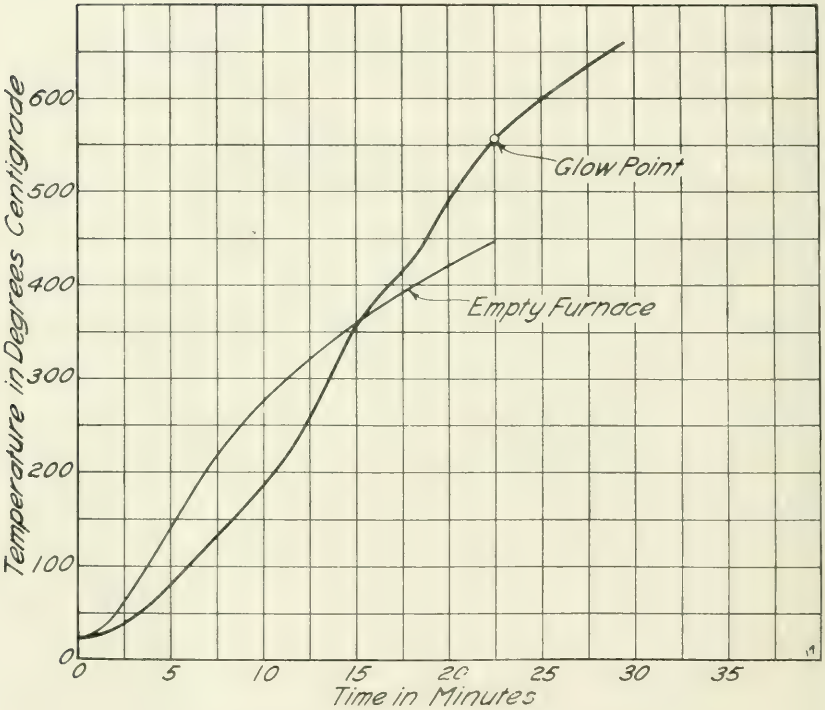

FIG. 17. HEATING CURVE, COAL NO. 8

Coal No. 9.

(Anthracite. Pennsylvania.)

The glow point of this coal was much higher than any of the bituminous coal or the lignite, as might be expected if the glow point is any criterion of ignition temperature. The heating curve as shown in Fig. 18 is an extremely smooth curve from the lower temperatures to the glow point, showing no exothermic reactions because of the relatively small amount of volatile matter present. As the heating of anthracite coal to the glow point was more a process of furnishing heat to a substance with a definite specific heat than heating something which contains certain volatile constituents requiring heat for their vaporization, its heating curve more nearly parallels that of the empty furnace. To be sure at a temperature of about 200 deg., anthracite began to furnish heat and its heating curve to gain on the furnace

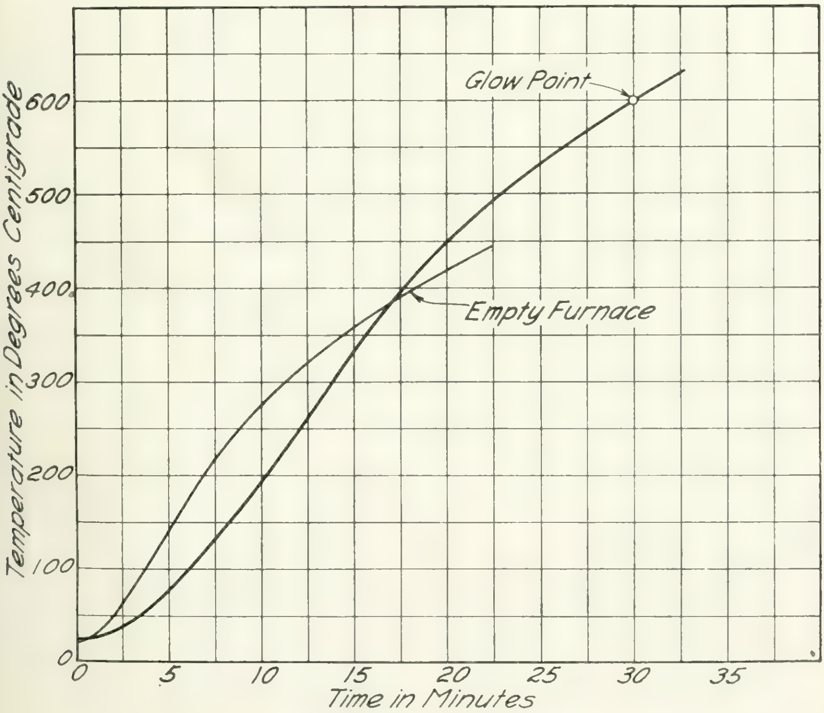

FIG. 18. HEATING CURVE, COAL NO. 9 (ANTHRACITE)

curve, but this gain is not as abrupt nor as well marked as in the case of the higher volatile substances illustrated in the previous diagrams. It is more apparent than ever in this heating curve and in that of coke, Fig. 19, that the glow point was merely an incident in this heating and did not mark a noticeable change in the heating rate.

Coke.

(Heating Curve, Fig. 19.)

This heating curve is even smoother and closer to the curve of the furnace than that for anthracite. However, at approximately the same temperature, 200 deg., self-heating began, causing a gain in the coke heating curve over the furnace curve.

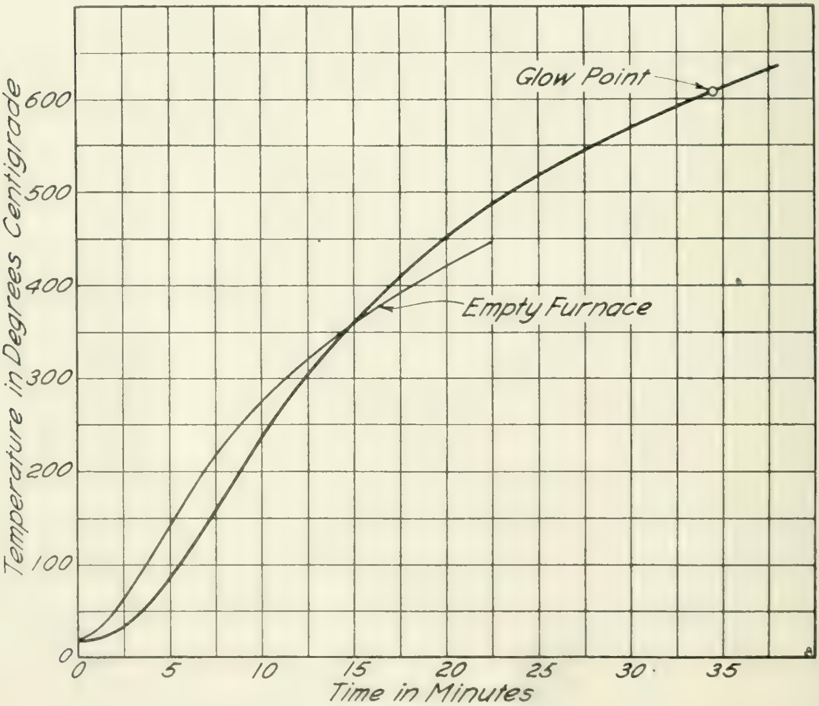

Fig. 19. Heating Curve, Coke

X. CONCLUSIONS.

32. *Summary.—*

(1) The "Ignition Temperature" of coal means nothing unless it is applied to some definite point in the process of heating the coal, which can easily be determined and duplicated.

(2) The temperature at which the coal glows seems to be the most logical point to choose as the ignition temperature. It is easily observed, can be duplicated with a fair degree of accuracy, and marks the beginning of visible combustion.

(3) The glow point is probably affected by the oxygen content of the coal, and perhaps by other agencies made active by weathering.

(4) The glow point is not affected by ash, moisture, size of particles, slight variations from the normal air supply, or rate of heating.

(5) There is no indication at present that the glow point bears any direct relation to the liability to fire while in storage. Perhaps if a series of tests were made on the glow points of freshly mined coal new information would be brought to light which would lead to the discovery of some more definite relation between glow point and the firing qualities of the coal.

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UNIVERSITY OF ILLINOIS
ENGINEERING EXPERIMENT STATION

BULLETIN NO. 129 MAY, 1922

AN INVESTIGATION OF THE PROPERTIES OF CHILLED IRON CAR WHEELS

PART I

WHEEL FIT AND STATIC LOAD STRAINS AND STRESSES

CONDUCTED BY
THE ENGINEERING EXPERIMENT STATION
UNIVERSITY OF ILLINOIS

IN COÖPERATION WITH
THE ASSOCIATION OF MANUFACTURERS OF
CHILLED IRON CAR WHEELS

BY
J. M. SNODGRASS
AND
F. H. GULDNER

ENGINEERING EXPERIMENT STATION
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AN INVESTIGATION OF THE PROPERTIES OF CHILLED IRON CAR WHEELS

PART I. WHEEL FIT AND STATIC LOAD STRAINS

I. Introduction

1. *Preliminary.*—The car wheel is one of the important factors which have made possible the great development of transportation facilities that has occurred within the past seventy years. Although during this time many types of car wheels have been utilized, only five have come into common use, thus demonstrating their general fitness to meet the severe service conditions now existing. These five types are the rolled steel wheel, the forged steel wheel, the cast-steel wheel, the built-up wheel, and the chilled iron wheel. The first three types are manufactured by processes and of materials indicated by their names. The built-up wheel, as the name suggests, consists of either a spoked wheel of cast-iron, or a disc, upon the periphery of which is attached a tire of rolled steel.

The chilled iron wheel is essentially a cast-iron wheel whose tread, or that part of the wheel which comes in contact with the track, consists of an extremely hard metal capable of resisting both excessive deformation and wear, while the remainder of the wheel consists of relatively soft and easily machined cast-iron. The hardened tread is produced by chilling the outer circumference or tread of the wheel when the metal is poured. A more detailed description of the chilled iron wheel, with such additional information regarding its manufacture and properties as may be of service in the interpretation of the results of this investigation, is presented in Appendix A, page 75. In service, the use of the built-up wheel is confined almost wholly to passenger coaches, locomotives, and tenders, whereas the rolled, forged, and cast-steel wheels, and the chilled iron wheel are in use under locomotive tenders, passenger coaches, and freight cars. For various reasons the use of steel wheels predominates in passenger service, while the relative cheapness of chilled iron wheels, together with certain other advantages they possess, makes their use almost universal for freight service.

Seventy years of transportation history show a very great increase in car capacity, and, as a consequence, greatly increased wheel loads both in freight and in passenger service; yet the same period of time does not show a proportionate increase in the size and presumably in the strength of the car wheels used, a fact which is especially true with relation to the chilled iron car wheel used in freight service. As an example, the wheel load in freight car service has increased from 5 000 lbs. on the 10-ton capacity car of 1870, to 25 000 lbs. on the 70-ton capacity car of the present time, an increase of 400 per cent. During the same time the weight of the wheel has been increased from 525 lbs. for the 5 000-lb wheel load to 850 lbs. for the 25 000-lb. wheel load, or an increase of 60 per cent. From comparisons of this kind it is evident that increase in wheel load has not been accompanied by a corresponding increase in the weight of the wheel. As the chilled wheel of today is carrying the imposed loads with a low percentage of failures, it seems evident that unduly large factors of safety may have been present in the earlier period. Furthermore, the conclusion may be drawn that, if the policy of increasing wheel loads continues without providing proportionate increase in wheel weight, a time will eventually arrive when the wheel no longer can withstand the imposed loadings.

Considering the relatively small amount of trouble that car wheels in general have given heretofore, and referring in particular to the chilled iron wheel, it is not surprising to find that but little experimental work has been done with regard to car wheel problems.* As a consequence, the stresses and strains in car wheels due to various service conditions have been largely a matter of speculation and conjecture, based mainly on the relatively small amount of information available from failures of wheels in service. By a careful analysis of these failures, railroads and wheel manufacturers, independently at first, then in small groups, and later in conjunction with the Master Car Builders Association, have, with comparatively slight changes in design and through moderate additions of metal, been able to produce wheels which have at all times satisfied service requirements.

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* Report of Tests to Determine the Stress in the Plate of Cast Iron Wheels Due to the Heat Produced by the Brake Shoe. L. E. Endsley. Proc. West. Ry. Club, p. 194, Mar. 27, 1914.

Standardization of Chilled Iron Crane Wheels. F. K. Vial. Proc. A. S. M. E., Dec. 1914.

Owing to the desirability of obtaining definite information concerning the magnitude and distribution of stresses in chilled car wheels and of determining the limitations of these wheels as used today, and with a further view of improving the chilled iron wheel in order to meet future requirements, a coöperative agreement was entered into by the ASSOCIATION OF MANUFACTURERS OF CHILLED CAR WHEELS and the UNIVERSITY OF ILLINOIS. Essentially the agreement provided that the Association furnish funds for all special equipment required and defray all expenses incident to carrying on the investigation, and that through its Engineering Experiment Station, the University conduct the research, provide the use of available laboratories, shops, and office facilities, and publish a report of the results. The principal items of the agreement are presented in Appendix C, page 94. After the approval of the agreement, committees under whose direction the work should be carried on were formed, and a conference was held in March, 1916, at which a tentative program was laid out.

2. *Scope of Investigation.*—The original program was revised and extended at various times until it has embraced tests to determine:

(1) the strains caused by mounting the wheel on its axle;
(2) the strains caused by the static or wheel loads;
(3) the ultimate breaking strength of flanges, and strains caused by flange pressure;
(4) the strains, due to temperature gradients in the wheel, caused by brake application;
(5) incidental problems related to the above.

The present bulletin gives the results of a series of strain-gage tests in connection with the items (1) and (2) just mentioned; that is, tests made to determine the strains produced within the wheel by mounting it on its axle, and by the application of wheel loads. The strain within the wheel caused by forcing it on an axle was first determined for two 33-in. 725-lb. M. C. B. wheels. The same pair of mounted wheels was then subjected to static loads ranging from 20 000 to 200 000 lbs. per wheel, and the resulting strains noted. The loading effect was produced by applying the load to the axle by means of a testing machine, and allowing the wheels to transmit it to a pair of rails, the conditions being similar to those found in service. A similar set of tests was carried out with a pair of 33-in. 740-lb. Arch Plate type of wheels. Additional tests for the purpose of obtaining more complete information concerning the mounting strains than was given by the above mentioned tests were then made upon a pair of 33-in. 725-lb. M. C. B. wheels. The reporting of the results of the mounting and static tests upon the wheels just mentioned is the purpose of the present bulletin. It is intended to publish additional reports dealing with tests to determine the ultimate strength of flanges, the effect of flange pressure, and the effect of brake applications, together with the related problems of brake friction and the thermal expansion of cast-iron.

3. *Acknowledgments.*—The Association of Manufacturers of Chilled Car Wheels appointed George W. Lyndon and F. K. Vial, President and Consulting Engineer, respectively, of the Association, as a committee to assist in promoting the investigation. In addition to the financial help given by the Association, the investigation owes much to the assistance rendered by this committee in the way of technical advice, helpful interest, and effective coöperation throughout the progress of the work. The Engineering Experiment Station of the University of Illinois appointed Edward C. Schmidt, Professor of Railway Engineering, and Arthur N. Talbot, Professor of Municipal and Sanitary Engineering, as a committee in charge of the investigation. Professor Schmidt withdrew from the University to enter the United States Military Service during the late war, and was replaced on this committee by John M. Snodgrass, Professor of Railway Mechanical Engineering. In the direction of the investigation this committee has had frequent conferences with the committee of the Association of Manufacturers of Chilled Car Wheels.

The greater part of the experimental work was carried on by F. H. Guldner, who became connected with the work a few months after its inception. R. E. Turley and O. S. Beyer, Jr., both of whom were connected with the experimental work during the earlier part of the investigation, did much in the way of developing methods, making some early tests, and getting the work under way. The committees are further indebted to H. F. Moore, Professor of Engineering Materials at the University of Illinois, for his assistance and advice upon various matters relating to the investigation.

II. PROBLEM OF CAR WHEEL UNDER LOAD

4. *General Statement Concerning Wheel Loading.*—A car wheel in service is subjected to conditions which produce stress within the wheel. These stress-producing conditions may exist in a great variety of combinations, and give rise to stresses of a more or less complex nature. The principal causes of stress in a car wheel are:

(1) manufacturing processes, which may cause initial stresses;
(2) forcing the wheel on the axle, or mounting;
(3) the proportion of the car loading supported by one wheel;
(4) the lateral pressure on the wheel flanges produced by rounding curves, by wind, or by the unevenness of the track;
(5) non-uniform temperatures in the wheel caused by brake application;
(6) centrifugal force, when the speed is high.

Rotation, moreover, complicates the problem still further by introducing impact and repeated stresses.

Important initial stresses, if existent, can be traced to improper manufacture. The process of annealing* has for its object the elimination of this type of stress, and, if it is properly performed, either removes the initial stresses entirely or reduces them to small magnitude. Concerning the magnitude and distribution of this type of stress, relatively few definite data exist.†

After being cast, the wheel is bored slightly smaller than its axle and forced upon it. In general, this mounting of the wheel on its axle produces compression in radial directions and tension in tangential directions throughout the entire wheel. The intensity of these stresses is greatest at the bore and decreases toward the tread until the intersection of the inner and outer plates is reached, at which point there may be a slight increase in magnitude, beyond which a decrease again occurs. At the rim or tread of the wheel these stresses become negligible.

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* See Appendix A, p. 45.

†Interstate Commerce Commission. Report of the Chief of the Bureau of Safety, Covering the Investigation of an Accident which Occurred on the New York Central Railroad near Waterloo, Indiana, March 21, 1917, pp. 8-10.

On being placed in service, the wheel doubtless encounters conditions that produce an exceedingly complex stress distribution within it. The static load, or proportional part of the total weight of the loaded car, is transmitted by the axle to the wheel and through it to the rail. As a consequence, tensile and compressive stresses of varying intensity are set up in all parts of the wheel. The compressive stress produced by the static load is at a maximum on the radial line connecting the center of the wheel and the point where the rail is in contact with the wheel. No general statement can be made as to where the actual maximum stress occurs on this radial line.

The order in which additional stresses occur in the wheel is necessarily determined by the conditions under which the wheel is operating. Lateral pressure on the wheel flange when rounding a curve adds to or modifies stresses already existing in the wheel. Both the static load and flange thrust stresses may be considerably augmented by impact, as in the case when heavily loaded cars operate on imperfectly aligned track, or strike guard rails and crossings at high speed. The application of the brakes for purposes of car retardation generates heat that must be dissipated by the brake shoe and the wheel. Within the wheel this dissipation of heat produces unequal temperatures or gradients between different points, with resultant stresses. If not properly handled by operating men, long continued brake application, such as occurs in mountainous regions, may occasion abnormal stresses and be a source of trouble, with attendant possibilities of serious disaster. Centrifugal force or a hot journal may play a part in causing undue stress within a wheel. Due to rotation, moreover, all the stresses present, excepting initial stresses and those due to mounting, occur as repeated stresses, thereby still further complicating the general problem.

Of the more important means of producing internal stress in the wheel it may be generally stated, with the exceptions subsequently noted, that the mounting of the wheel and the static load tend to produce compressive stresses in a radial direction, which stresses may be wholly or partially relieved or even reversed by tensile stresses due to centrifugal force and brake application. On the outer face of the wheel the stresses due to flange thrust are cumulative with, while on the inner face they are opposed to, those caused by centrifugal force and brake application.

Throughout the investigation the attempt has been made to simulate service conditions in the methods used in loading the wheels in so far as the laboratory facilities would permit. In addition to this, most of the wheels tested were subjected to conditions exceeding in severity those which would result from normal wheel service. The wheels tested were for the most part taken at random from the stock of the Griffin Wheel Company, and are assumed to be representative of their respective types.

5. *Evaluation of Stresses Existing in a Loaded Car Wheel.*—An examination of the section of a chilled car wheel indicates that, for purposes of analysis, it might be considered as a series of curved plates in which the outer edge of one becomes the inner edge of a succeeding one, or vice versa; furthermore, that the stresses existent in any individual section would be a function of those occurring in adjacent sections. Attempts have been made to derive the theory underlying curved plates, but as yet no satisfactory analysis has been made. Until such an analysis is available it will probably be impossible to predict the intensity or distribution of stresses in a car wheel by analytical methods. Hence, when it becomes either necessary or desirable to know the conditions of stress in a shape as complex as that of a car wheel, an experimental method aids, although it does not completely succeed, in solving the problem.

From the foregoing it can be seen that the general problem is a complex one and that, due to the irregular shape of the wheel, it cannot at present be satisfactorily analyzed by theoretical methods. Accordingly the attempt has been made to determine, as far as is possible by experimental methods, the individual effects caused by the principal loads producing stress in a car wheel, and in several cases to determine their cumulative or combined effect. Having found the individual effects, combinations can then be estimated for any assumed operating condition with greater precision than has heretofore been possible.

It is evident that the problem of the car wheel under load is one of compound stress; that is, stress in more than one direction. Throughout this report the term stress has been used to indicate the stress which would exist in the material under observation if subjected to either simple tension or simple compression, and, under these conditions, deformed to an extent equal to the measured strain. The strains measured in these investigations constitute the fundamental data taken, and these measured strains are for the most part due to the effect of more than one stress acting within the material; that is, they are due to compound stress. Concerning compound stress there are three theories more or less generally accepted, none of which, however, can be considered wholly acceptable in the determination of the actual stresses existing within cast-iron subjected to such loads as car wheels necessarily carry. These three theories are known respectively as the maximum shear, the maximum stress, and the maximum strain theory.* It seems safe to assume that the maximum shear theory does not apply to cast-iron, and should not be used in connection with the problem in hand. The application of the maximum stress theory would in general lead to a lower evaluation of the existing stresses than would an application of the maximum strain theory. The computed stresses which are presented and discussed throughout the report are determined from the measured strains and a stress-strain relation determined from specimens tested under simple tension and simple compression. As reported, therefore, they are not in general the stresses which exist in the wheel, and may not be used as a measure of the resistance developed in any section in the way that a simple stress is used. The values of the stress are given to allow the reader to use these approximate values of a more familiar concept than strain. The word stress has throughout the discussion frequently been qualified by the use of the expression ''corresponding simple'' when referring to the stresses computed from the measured strain and the stress-strain relation used; and this expression has been used to call attention to, and to emphasize the fact that the stresses under consideration are for the most part computed from measured strains resulting from compound stress and from a stress-strain relation based upon simple tension and compression.

As the available information concerning the theories does not appear to indicate definitely that any of them, either separately or in combination, should be applied in the case of cast-iron, it is felt desirable that the matter of the actual severity of these stresses should be considered as held in abeyance at this time. The report, moreover, is not intended to present figures which can be taken as exact values of the actual stresses existing where the strains from which the values given were calculated are produced by compounded stress. No attempt has been made to report exact values for the stresses existing in the wheels under conditions of actual service, as this report is concerned only with the stresses produced through two kinds of wheel loading, namely, mounting and static loading. The stresses resulting from these two forms of loading may be materially modified by additional stress-producing factors to which a wheel may be subjected in service. The usual stress condition produced in the parts of the car wheel by mounting, and through the application of static loads, is that of material subjected to tension in one direction and compression in a direction at right angles thereto. This condition produces a deformation or strain in the direction of either stress, which is greater than would be produced by the stress in that direction acting alone. Any estimates regarding factors of safety, based upon the stresses reported for the mounting and static load tests, and considered as applying to the chilled iron wheel or its parts, could therefore be properly considered as conservatively estimated; that is, if the actual stresses could be determined, it is probable that this would not involve a reduction in the estimated values of the factors of safety, but would rather justify increased values for them. In this report no attempt has been made to indicate values for factors of safety.

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* For a brief discussion of these theories see, "The Strength and Stiffness of Steel Under Biaxial Loading." Univ. of Ill. Eng. Exp. Sta., Bul. 85, 1915.

III. Physical Properties of Chilled Car Wheel Irons

6. *Selection and Treatment of Specimens.*—In order to assist in interpreting the data obtained in the tests made on the car wheels, a number of specimens of wheel irons were taken, both from wheels tested, and as coupons from the foundry ladle. Some of the properties of these specimens which were of most importance to the investigation were determined.

In general, in discussions pertaining to the strength of materials it is customary to consider the matter in terms of stress, where stress is that internal force which, when a body is subjected to external forces, tends to hold the molecules in their original relation and to preserve the integrity of the body. The fundamental data obtained in these tests were, however, strains or deformation measurements. Hence, to facilitate interpretation of the strain data in terms of stress, it became necessary to have a knowledge of the stress-stain relation of the metal used in car wheels. This relation was determined by applying a known load to the metal under test, and measuring the corresponding elongation or contraction; then, by calculation from the load and the deformation, were obtained the stress per unit area and the strain per unit length, respectively. Measurement of the strain, along with a knowledge of the relation between stress and strain, permitted an evaluation of the stress produced. Accordingly the stress-strain data, and in addition the ultimate strength and modulus of elasticity, were determined for the car wheel and coupon specimens obtained. Since the stress-strain relation is dependent largely upon the composition of the metal, chemical analyses were made to determine total carbon, combined carbon, silicon, manganese, phosphorous, and sulphur. Hardness tests, both by the Brinell method and with the scleroscope, were also made in an attempt to associate the quality of hardness with other properties of the metal. The origin, shape, and treatment of the specimens, together with the results of the chemical and physical tests, are shown in Table 1 and in Figs. 1 to 5 inclusive.

On account of the irregular shape of the wheel (see Fig. 1, *a* and *b*) in the region between the hub and the inner side of the tread, it was difficult to remove specimens of any considerable length. For this reason the specimens removed from the wheel and used for the

TABLE 1

PHYSICAL AND CHEMICAL PROPERTIES OF CAR WHEEL IRONS AND COUPONS

Specimen Number	Wheel Number	From	Chemical Analysis						Hardness		Stress-Strain			Remarks
			CC	Total C	Si	Mn	P	S	Brinell	Shore	Mod of Elas at 5000 lb per sq in in Millions	Ult. Strength lb per sq in	Kind of Test	
B2		33-725" Wheel	041	371	055	063	0364	0213	137	37.9	19	28000	Tensile	See Figs 1a, 2a, & 3
D1			067	3.52	057	063	0.350	0214	143	358	18	26300	"	"
D2		"	030	366	055	063	0362	0215	130	36.1	22	27700	"	"
D3		"	048	373	056	062	0337	0214	128	380	20	27700	"	"
D4		"	057	366	056	065	0354	0222	140	382	28	26900	"	"
D5		"	0.50	363	059	064	0350	0.216	133	348	19	26300	"	"
D6		"	082	368	0.58	063	0350	0215	144	37.4	17	24200	"	"
F1		"	0.61	370	056	065	0.351	0222	141	35.2	16	28000	"	"
F2		"	047	3.75	059	063	0363	0219	134	35.6	22	26600	"	"
F3		"	047	371	0.60	0.63	0359	0.221	110	344	17	24900	"	"
F4		"	058	366	056	0.65	0340	0224	138	35.4	15	26400	"	"
F5		"	051	379	060	063	0.351	0216	122	35.4	18	26400	"	"
F6		"									17	23300	"	"
B4		"										29500	"	"
B5		"										28600	"	"
B6		"										25200	"	"
E1		"	060	3.59	057	064	0350	0214	120	30.0	17	62500	Compressive	See Figs 1a, 2b, & 5
E2		"	040	368	0.59	0.61	0363	0.213	131	37.0	21	75000		"
E3		"	0.41	375	060	064	0.373	0.216	136	370	17	74200	"	"
E4		"	061	3.74	061	062	0338	0220	144	398	25	77700	"	"
E5		"	052	3.60	0.59	0.64	0.360	0215	138	35.5	20	75200	"	"
E6		"	0.59	365	0.59	0.64	0.350	0215	151	401	17	75000	"	"
C1		"	075	3.63	060	067	0.347	0.216	142	398	15	77400	"	"
C2		"	041	3.70	0.58	063	0369	0212	128	348	16	73900	"	"
C3		"	050	3.69	0.55	059	0.345	0204	130	35.0	19	73200	"	"
C4		"	070	3.71	0.56	0.67	0.354	0.227	137	36.4	19	82900	"	"
C5		"	046	375	0.57	064	0350	0.218	130	35.1	19	72900	"	"
C6		"	074	377	0.58	063	0368	0206	128	358	17	74200	"	"
A1	16129	33-790" Wheel	120	364	061	047	0340	0224			18	26000	Tensile	See Figs. 1b, 2c, & 3
A2	"	"	"	"	"	"	"	"			15	27100	"	"
A3	"	"	"	"	"	"	"	"			19		"	"
B1	22121	"	1.12	3.53	072	056	0.340	0252			16	32600	"	"
B2	"	"	"	"	"	"	"	"			18	32800	"	"
B3	"	"	"	"	"	"	"	"			19	32700	"	"
1PP		Coupon	1.08	375	0.57	060	0.330	0.197			16	26200	"	See Figs. 2d & 3
3PP		"	0.86	3.64	1.44	0.63	0.330	0.176			14	26500	"	"
1	51092	"	153	382	054	038	0256	0.174	164	249	24	24500	"	Sand Cooled, See Figs. 2c & 4
1	"	"	1.54	382	052	039	0.259	0.175	150	25.3	21	22700	"	" "
1A	"	"	1.28	3.86	053	038	0.250	0.176	190	32.6	18	23300	"	Wheel & Coupons Pitted Std. Lgth Time
1A	"	"	130	379	0.51	0.39	0253	0.174	184	31.3	21	24000	"	"
2	51091	"	113	3.88	0.53	0.38	0257	0.176	242	40.2	25	32200	"	Sand Cooled
2A	"	"	113	382	0.51	039	0.266	0.176	207	372	19	24900	"	Wheel & Coupons Taken from Molds &
2A	"	"	130	384	051	0.39	0.257	0175	242	410	18	33200	"	Cooled with Water Wheel Broke
3	51090	"	Trace	385	0.52	039	0252	0187	91	188	16	15200	"	Sand Cooled
3	"	"	"	3.88	0.53	038	0258	0.190	89	184	20	15400	"	"
3A	"	"	1.36	377	0.51	040	0254	0178	204	332	18	23800	"	Wheel & Coupons Pitted Triple Std Time
1A1		"	123	382	050	058	0180	0130			16	25500	"	See Figs 2c & 4 ←
2A1		"	084	379	0.57	054	0190	0.132			17	24400	"	"
3A3		"	076	379	066	055	0.220	0128			17	23400	"	"
4A1		"	0.82	379	0.58	0.63	0210	0186			19	24500	"	"
5A1		"	072	382	063	065	0.200	0170			15	22600	"	"
6A1		"	0.79	3.81	0.55	0.56	0190	0143			14	20800	"	"
1B2		"	080	369	0.55	0.59	0.230	0.124			16	27400	"	"
2B1		"	0.90	368	061	067	0.240	0.163			18	26000	"	"
3B2		"	089	368	074	0.60	0.260	0138			15	24200	"	"
3C1		"	087	3.77	079	048	0.360	0164			17	27100	"	"
L3		"	074	3.52	0.64	061	0200	0.151			22	31800	"	"
L4		"	1.04	3.32	0.63	0.59	0.320	0.249			28	31600	"	"

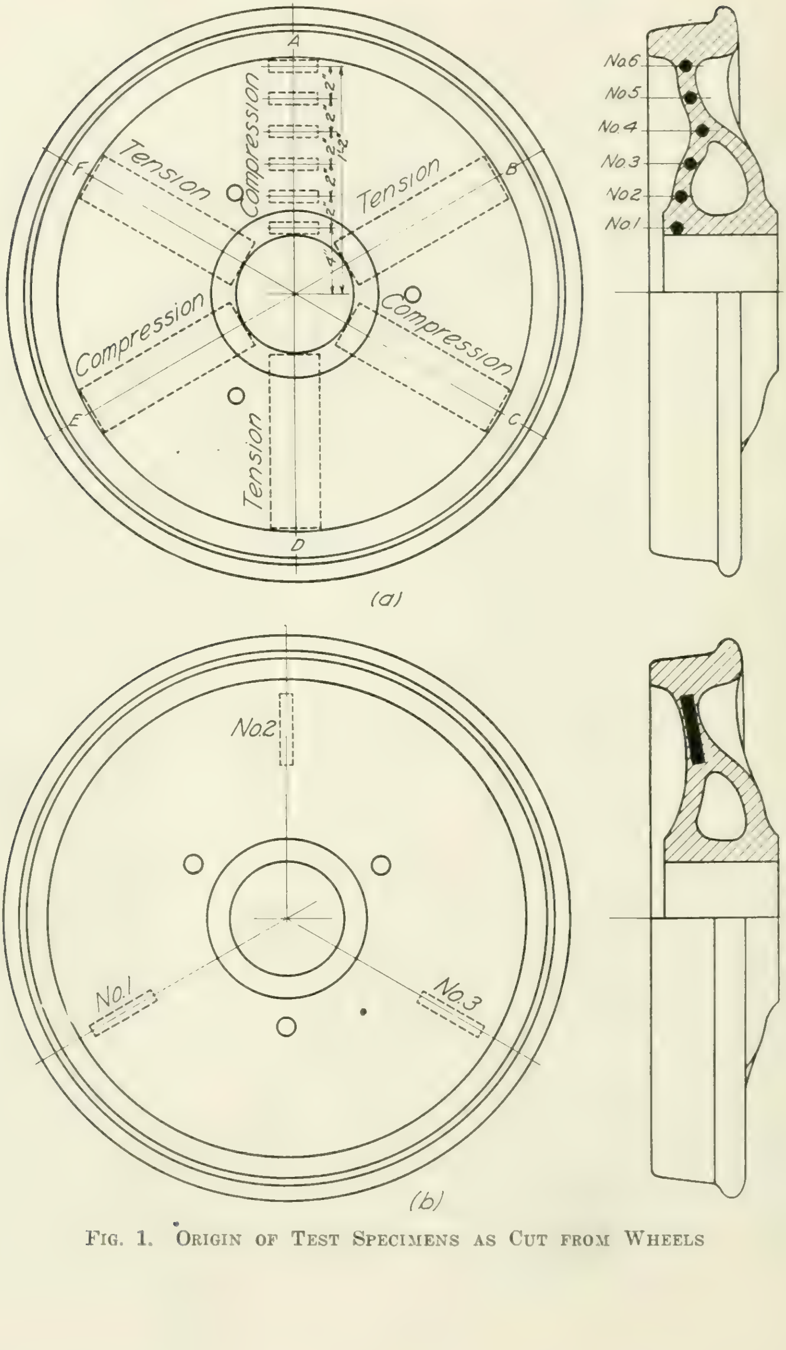
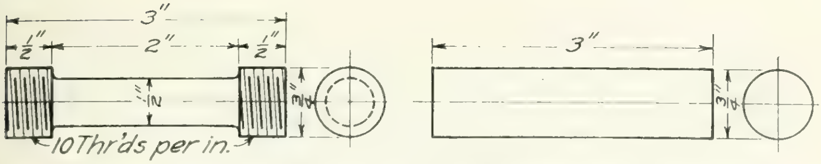
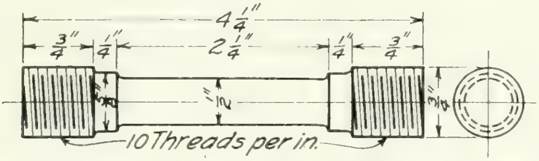


FIG. 1. ORIGIN OF TEST SPECIMENS AS CUT FROM WHEELS

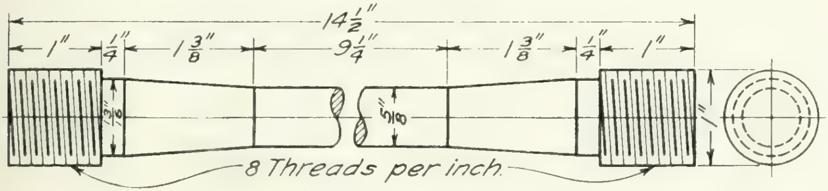


(a)-Tension Specimen from 33 in., 725 lb. Wheel. 1.25 in. Gage Length.

(b)-Compression Specimen, from 33 in., 725 lb Wheel. 1.25 in. Gage Length.



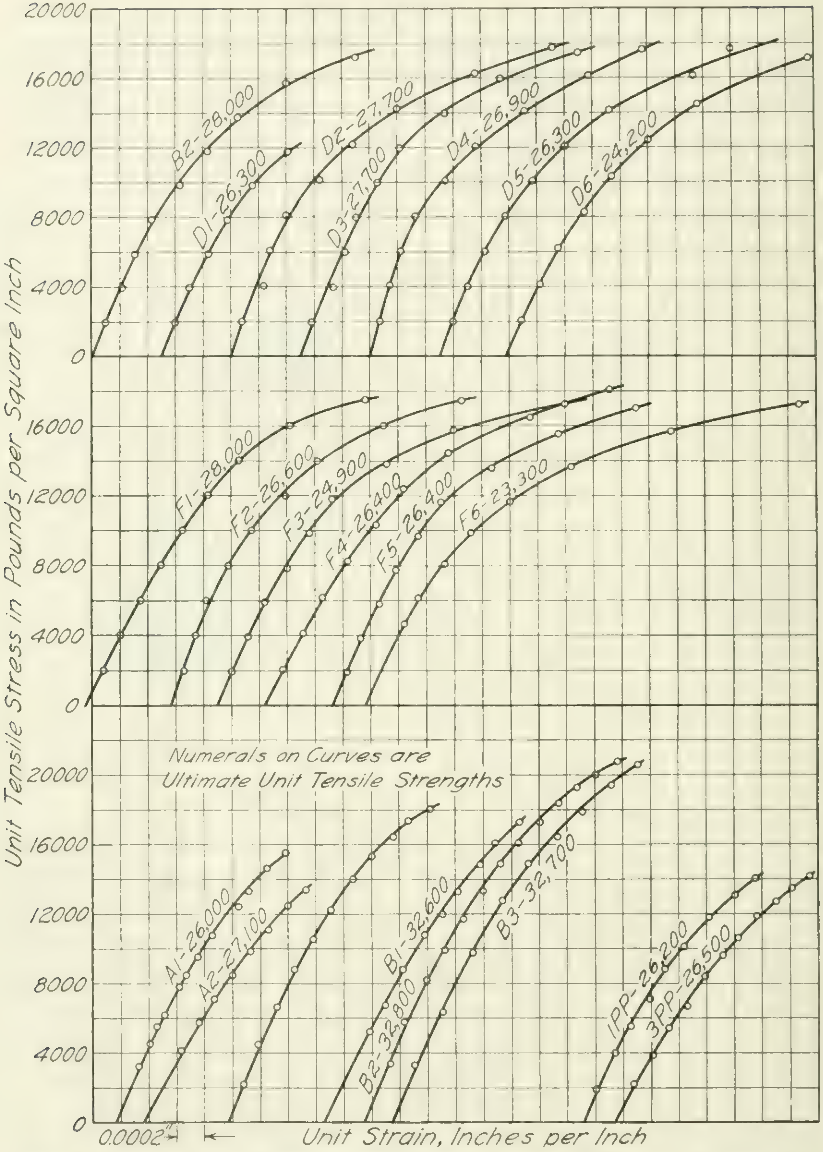
(c)-Tension Specimen from Wheels No. 16129 and No. 22121 and from Coupons. 2.0 in. Gage Length.



(d)-Tension Specimen.

Fig. 2. Details of Test Specimens

tension tests did not conform to the dimensions recommended by the American Society for Testing Materials. The treatment to which the specimens were subjected varied. In the case of specimens taken from the wheel, the removal was made after the wheel had been cooled or annealed, as shown in the table. The coupons were cast in sand molds, and were either allowed to cool in the sand, or. after solidification, were placed in the pit with the wheel they represented. The number of specimens already tested is small, but it is hoped that in time a sufficient number of like specimens may be subjected to tests so that a more complete study of the physical properties of car wheel irons may be made.



Numerals on Curves are Ultimate Unit Tensile Strengths

FIG. 3. STRESS-STRAIN RELATION FOR WHEEL IRONS AND COUPONS (TENSION)

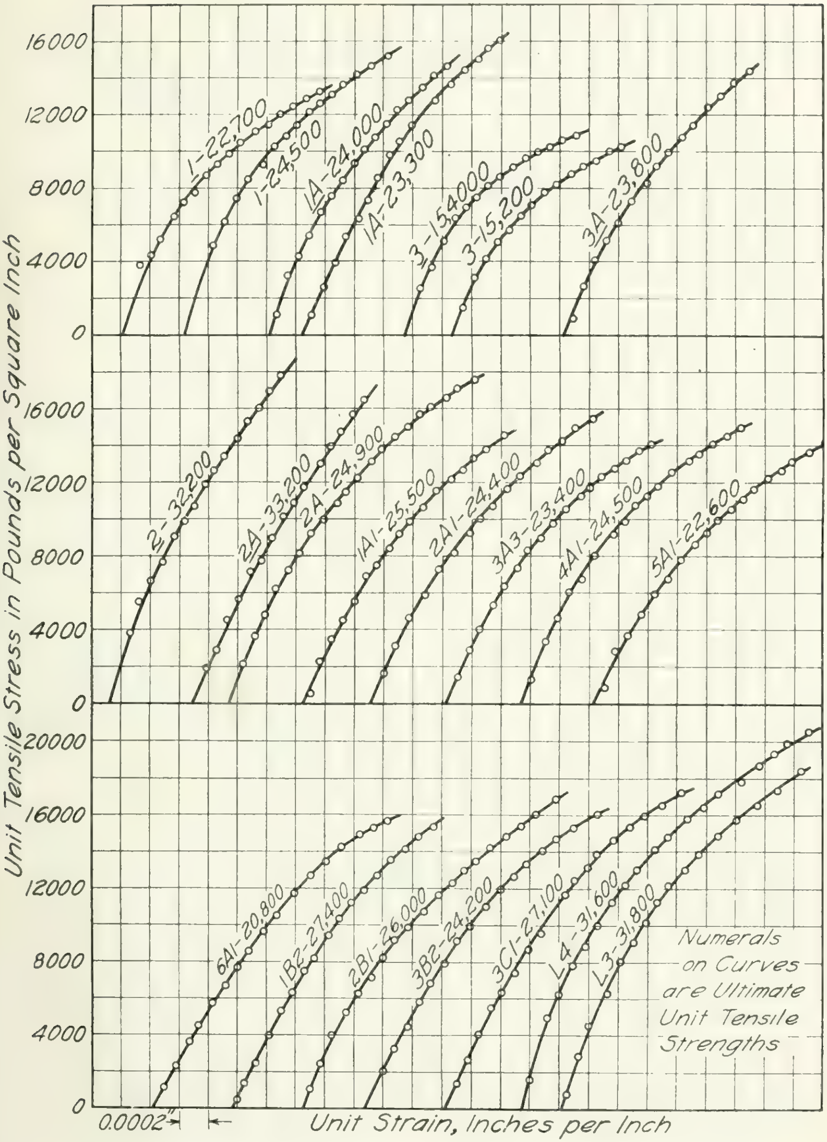


FIG. 4. STRESS-STRAIN RELATION FOR WHEEL IRON COUPONS (TENSION)

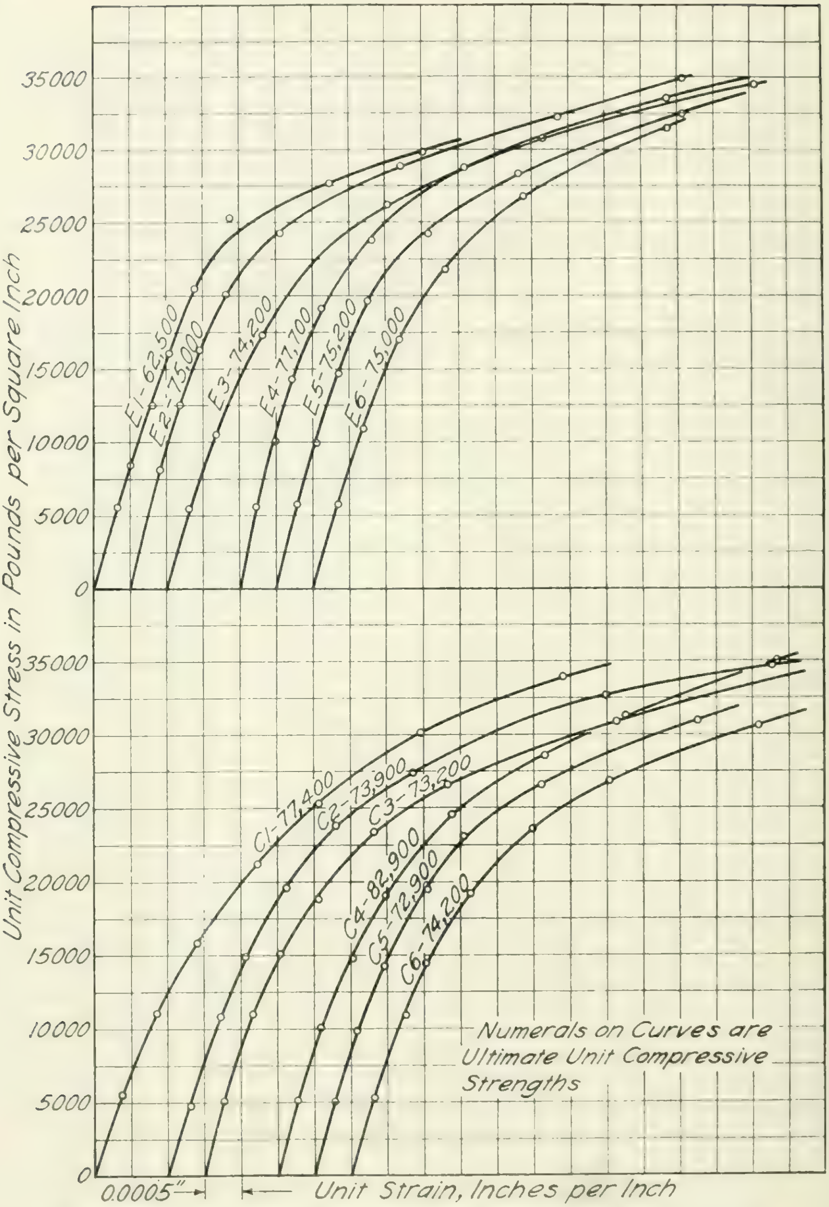


FIG. 5. STRESS-STRAIN RELATION FOR WHEEL IRONS (COMPRESSION)

7. *Chemical Analyses.*—The chemical analyses were made in the laboratory of the Griffin Wheel Company, Chicago. The amounts of the various elements present in the specimens are shown in Table 1 and are expressed in per cent.

8. *Stress-Strain Relation.*—For these determinations a 100 000-lb. Riehle Universal Testing machine, using spherical seated holders or bearing blocks, was used. The machine was stopped for readings.

A Ewing extensometer reading directly to 0.000071 in. per in., made by the Cambridge Instrument Company, was used in measuring the strains on the specimens shown in Fig. 2, *a* and *b*. For the 2-in. specimens an extensometer reading directly to 0.000025 in. per in. was used, and for the 8-in. specimens an instrument reading directly to 0.000062 in. per in.* The modulus of cast-iron being a variable, it was thought that the secant modulus† corresponding to a stress of 5 000 lbs. per sq. in. would permit suitable comparison, and the moduli are thus reported.

As no two of the specimens, even when cut from the same wheel, gave identical stress-strain relations, and as it would obviously be impractical to cut a test specimen from each of the positions in the wheel at which the strains were measured, it was necessary, in order to evaluate the stress from the strains as measured on the car wheel, to deduce two stress-strain curves which might be assumed to be representative of wheel iron in tension and in compression, respectively. For this purpose, the results for the 13 tension specimens, B_1 to F_6 inclusive, (see Table 1) were averaged, and from the mean a curve was drawn. In a similar way the mean compression stress-strain curve was obtained from the 12 compression specimens, E_1 to C_6 inclusive. As stated above, these two curves, Fig. 6, were assumed to show, respectively, representative tensile and compressive stress-strain relations for chilled wheel irons, and are the curves upon which the stresses hereafter given are based.

9. *Hardness.*—The hardness by the scleroscope method was determined with a Shore scleroscope made by the Shore Instrument

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* An illustration of the latter two instruments is shown in "The Relation between the Elastic Strength of Steel in Tension Compression and Shear." Univ. of Ill. Eng. Exp. Sta., Bul. 115, Fig. 4, p. 12, 1919.

† *Secant modulus* is defined as the slope of a straight line connecting the origin of the stress-strain curve and some arbitrarily chosen point on the curve, the curve itself being plotted with unit stress as ordinates and unit strains as abscissæ.

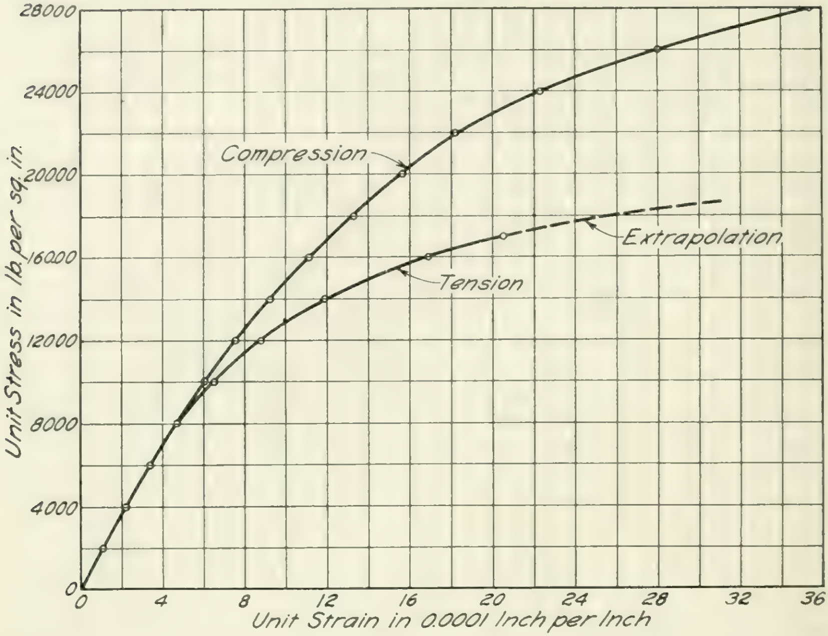

FIG. 6. ASSUMED MEAN STRESS-STRAIN RELATION FOR CHILLED WHEEL IRONS USED IN EVALUATION OF STRESS

and Manufacturing Company. This instrument measures the hardness by the measured rebound of a small weight carrying a blunt diamond which is dropped from a fixed height (vertically) on the material. The scleroscope data given in Table 1 represent the mean value of 10 readings taken on each specimen. The apparatus used for the Brinell tests was manufactured by Aktiebolaget Alpha, of Stockholm, Sweden. This test consists of forcing, under a definite pressure, a hardened steel ball of definite size into a flat plate of the material to be tested, and measuring the diameter of the indentation. The Brinell hardness is a function of the diameter of the indentation thus made. In the case of the tabulated Brinell data, they are the average of 3 or 4 observations using a 12-mm. ball with a pressure of 3000 kg. for 30 sec. The average hardness shown by the 36 samples cut from the wheel, Fig. 1a, at various distances from the center of the wheel is given in Fig. 7.

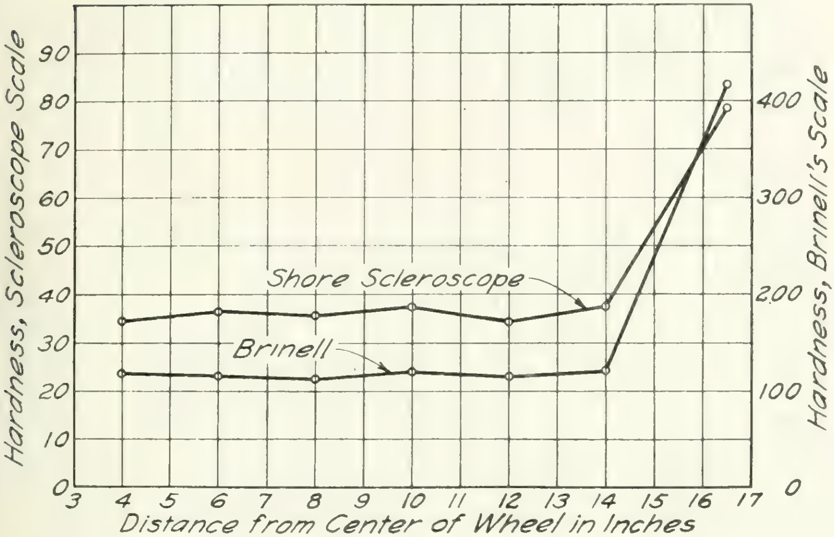

FIG. 7. MEAN HARDNESS OF 33-IN. 725-LB. CHILLED WHEEL AT VARIOUS RADII

10. *General Remarks Concerning Properties of Chilled Wheel Irons.*—It is recognized that proper application of chemical knowledge in the work of the wheel foundry has been an important factor in enabling the chilled iron wheel to meet the demands of modern railway operation. As a result car wheel manufacturers maintain chemical laboratories and control the chemical composition of wheels within fairly well defined limits. The railroads, however, with a few exceptions, do not embody chemical control in their specifications, but rely almost wholly on the results of physical tests in the acceptance of wheels.* Inspection of Table 1 shows that the chemical composition varies at different positions within the wheel. For instance in Table 1 the specimens, B_2 to C_6 inclusive, as removed from the plate of the wheel shown in Fig. 1a, show the following range in the percentage of the several elements:

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* Specifications for Cast-iron Car Wheels. Proc. M. C. B. Assn., Vol. 52, p. 490. 1918.

	Per Cent
Total Carbon	3.52 to 3.79
Combined Carbon	0.30 to 0.82
Silicon	0.55 to 0.61
Manganese	0.59 to 0.67
Phosphorous	0.337 to 0.373
Sulphur	0.204 to 0.227

With respect to the importance or effect of these variations no conclusions have been reached. For the same specimens the Brinell hardness varied from 110 to 151, while the scleroscope hardness ranged from 30.0 to 40.1. The secant modulus of elasticity of the several specimens had a minimum value of 15 000 000, whereas the maximum value was 28 000 000 lb. per sq. in. An appreciable variation from 23 300 to 29 500 lb. per sq. in. likewise existed in the ultimate strength of the specimens when they were tested in tension. In compression, the ultimate strength ranged from 62 500 to 82 900 lb. per sq. in. In the three specimens cut from wheel No. 22 121, the average ultimate tensile strength, 32 700 lb. per sq. in., is approximately 23 per cent greater than the average, 26 620 lb. per sq. in., found in the wheel represented by specimens B_2 to B_6 inclusive. Considering the important effect that tension plays in the failure of cast-iron, it would seem desirable to produce wheels having a high ultimate strength in tension, that is, having the property of toughness, provided, of course, this could be done without materially affecting the hardness, or the wearing qualities of the metal in contact with the rail.

If all the specimens given in Table 1 are considered, a variation of 100 per cent between the minimum and maximum moduli of elasticity in tension is seen to exist. A similar condition also exists with respect to the values of ultimate strength which are reported. The compression data likewise show considerable variation in the moduli and in the ultimate strengths recorded. This suggests that the metallurgy of wheel irons still offers a fertile field for improvement in quality, in order that wheels may meet the probable demands of future railway service.

No distinct relation was found between the ultimate tensile strength and the Brinell or scleroscope hardness, nor could a constant relation be determined between the Brinell and scleroscope hardness. This latter fact is readily apparent from inspection of Fig. 7. Over the distance 4 to 14 in. from the center of the wheel, the ratio of Brinell to scleroscope hardness is approximately 115 to 36 or 3.2, whereas at a radius of 16.5 in., that is, on the tread of the wheel, this ratio is 5.3. This figure also strikingly indicates that the extreme hardness produced by chilling is confined to the tread, and that the metal in the plate is relatively soft and of nearly uniform hardness.

IV. Apparatus and Methods Used in Mounting and Static Load Tests

11. *The Strain-Gage, and Its Use in Measuring Strains in Car Wheels.*—In the determination of strains developed by the various methods of stressing a car wheel, measurements were taken thereon in both radial and tangential directions with a Berry Strain-Gage illustrated in Fig. 8.

This gage consists of a frame carrying a fixed cone-shaped point, a multiplying lever, and an indicating dial. The short arm of the multiplying lever also carries a cone-shaped point, and the long arm of the lever actuates the plunger of the dial. This instrument measures the change of the distance between two small holes. These holes are the extremities of a gage-line along which the strain is desired. In making observations with the strain-gage, the cone points are inserted in the gage holes and the dial readings noted for each of the gage-lines on the wheel. These readings are first taken with the wheel in an unstrained condition. Another series of observations is then taken on the same gage-lines with known loads on the wheel. The difference between the initial and final readings divided by the product of the gage length and the multiplication ratio (as obtained by calibration) of the strain-gage lever, gives the actual unit strain for the load in question.

As a formula this expression becomes:

$$e = \frac{R_i - R_f}{l \times m}$$

where e = unit strain;
R_i = initial reading of the dial;
R_f = final reading of the dial;
l = length of gage-line;
m = multiplication ratio of strain gage.

The relative values of R_i and R_f, together with a consideration of the characteristics of the strain-gage used, indicate whether the strain is tensile or compressive. From the unit strain thus found and by reference to the stress-strain diagrams, Fig. 6, an approximate value of the corresponding simple stress, variously called the "true

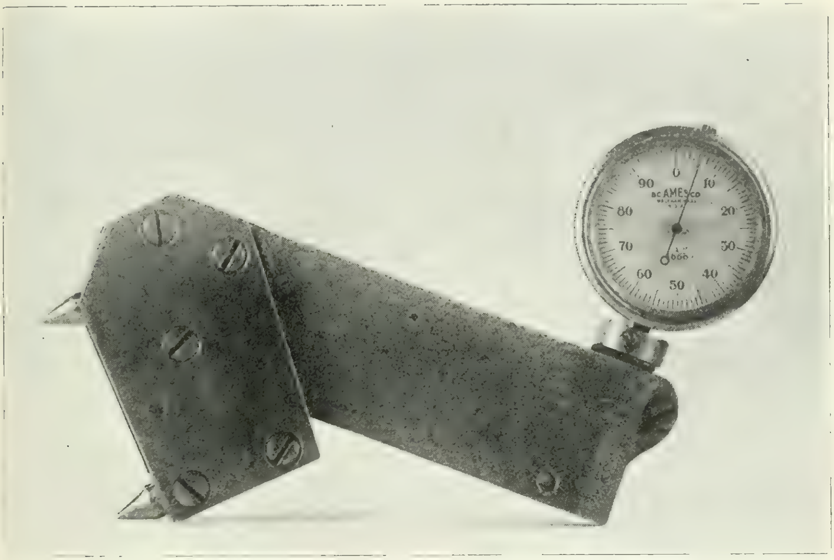

Fig. 8. Berry Strain-Gage

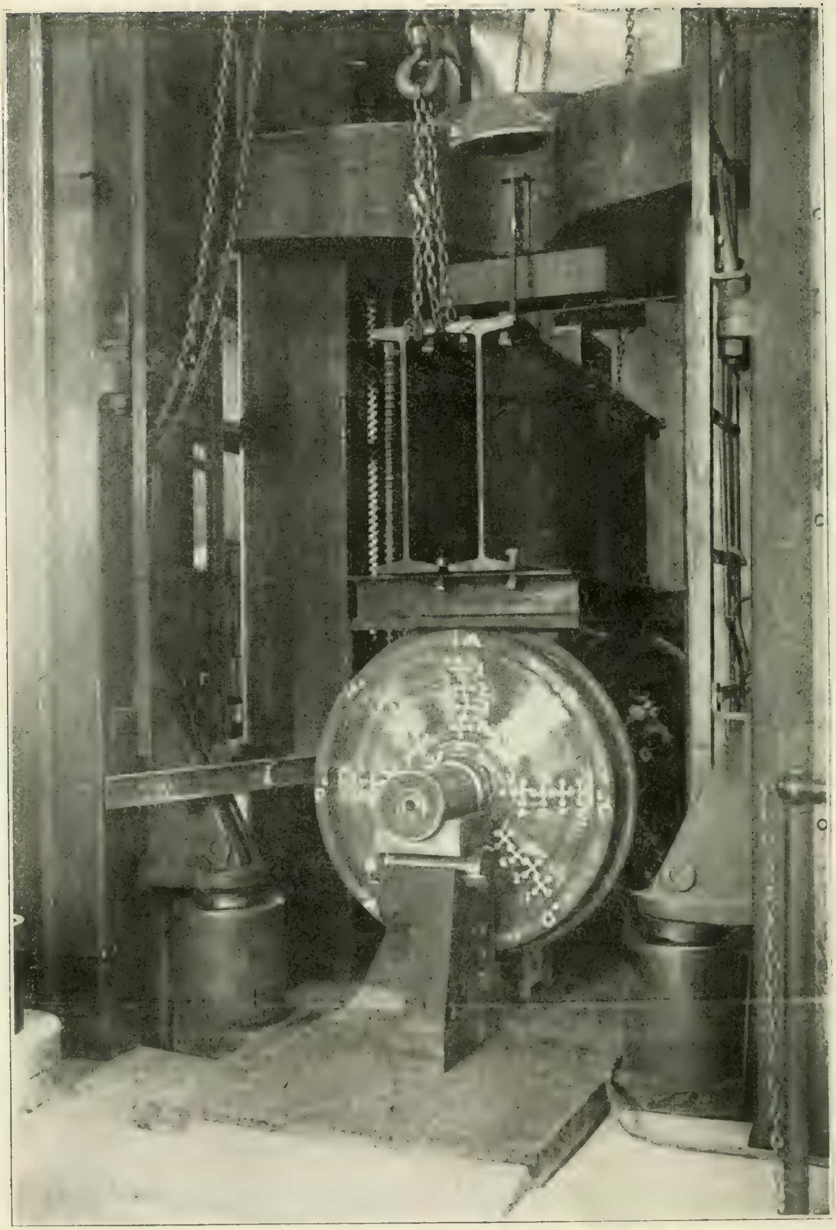

FIG. 9. 600 000-LB. TESTING MACHINE WITH PAIR OF WHEELS IN POSITION TO RECEIVE STATIC LOAD

stress,'' ''reduced stress,'' or ''ideal stress,'' is determined. Throughout this report any stress mentioned is in terms of the corresponding simple stress, or the stress that would exist if the metal were subjected to simple tension or compression and deformed to a degree equal to the measured strains. Both the initial and final strain-gage observations were taken in terms of a standard gage-line called a standard bar. This bar was made of the same material, in this case cast-iron, as that in which the strain was desired, and its use provided a means of compensating for changes in temperature of the material being tested and progressive changes due to other causes. It also furnished a check for noting any changes that might occur in the relative position of the dial and frame of the strain-gage. In these tests the gage-lines were established in both radial and tangential directions on the wheel by drilling holes 0.055 in. in diameter, approximately 1/16 in. in depth, and 2 in. between centers. . The instrument used had a dial graduated to 0.001 in. and a multiplication ratio of five. Hence the unit strain could be read directly to 0.0001 in. per inch, and by estimation to 1/10 of this figure.

Throughout the report certain gage-lines are shown for which no strains are given. For these the readings were not taken. The omissions are due to the fact that the intial strain-gage readings, i. e., those taken with the wheel in an unstrained condition, could not be duplicated when several trials were made. This was due to slight imperfections in the gage holes, to scale on the metal, etc. Since all the measured strains are dependent on these initial readings, the unreliable readings were discarded and no further observations were made on these gage-lines.

12. *Preparation and Methods of Testing Wheels for Determination of Strains Due to Mounting and Static Loads.*—A pair of wheels was prepared with gage holes determining lines in both radial and tangential directions and on several radii. With the strain-gage, readings were taken on each of the gage-lines before mounting the wheels on the axle. The two wheels were then pressed upon the seats of a standard 5½-in. by 10-in. M. C. B. axle in a 600 000-lb. Riehle testing machine. Autographic diagrams of the pressure during mounting were taken. After mounting, observations on each of the gage-lines were again made, and, as stated above, the differences, after proper

corrections, between the latter readings and the initial readings are measures of the unit strains.

After the strains incident to mounting had been determined, the wheels were placed in the testing machine, as shown in Fig. 9, and subjected to various known static loads.

For each of the static loads thus applied a complete series of strain readings was taken. In this case the difference between the initial readings, the observations taken before mounting, and the observations at the load in question, is the total strain in the wheel caused by the combined effects of mounting and the static load. It will be noted that, except for the inversion of the wheel and rail, the apparatus arranged for the static tests loads the wheel in the same manner as in actual service.

V. Results of Mounting and Static Tests

13. *Corresponding Simple Stresses in Two 33-in. 725-lb. M. C. B. Wheels Due to Mounting Wheels on Axle, together with Combined Effects of Mounting and Static Loads.*—The location of the gage-lines on these two wheels is shown in Figs. 10 and 11 and the gage-lines are designated as follows: the radial lines on which the gage-lines are located are lettered *A* to *H*, and these letters become the first figure in the designation; the relative position of the gage-line on the radius is next indicated by a numeral, which in turn is followed by either the letter *R* or *T*, signifying respectively a radial or a tangential gage-line; thus *B4R* denotes the gage-line or radial *B*, fourth from the tread and in a radial direction, and *B4T* indicates the gage-line similarly located but in the tangential direction.

The pressures required to mount the wheels on the axle together with the fit allowance are shown in Fig. 12, while the equivalent simple stresses resulting therefrom are given in Figs. 13 and 14.

Diagrams similar to Fig. 12 are regularly taken by both the manufacturers and railroads when mounting wheels. They serve as a check against placing improperly mounted wheels in service. With wheels properly fitted, this curve will show a nearly uniform increase in the pressure required for forcing the wheel on the axle from the instant the axle enters the wheel up to its final position. In addition, the final pressure as recorded indicates whether the recommendation of the M. C. B. Association in this respect has been fulfilled. If this pressure is much below that recommended, there is a possibility of the wheel becoming loose on the axle; whereas, if it is in excess of that recommended, cracked wheels may result. In view of these facts considerable attention is generally given to the final pressure required. For this type of wheel the M. C. B. Association specifies a final pressure ranging from 45 to 65 tons. The pressures recorded in the tests were 45.8 and 61.5 tons, or within the allowable range. Attention, however, is called to the fact that for a wheel seat diameter of 7 in. and a hub length equal to 7 in., the manufacturers' practice is to make the fit allowance 0.011 to 0.017 in., which results in a range of mounting pressures of about 45 to 65 tons. It will be noticed that the allowance on wheel No. 671237 was 0.021 in., and the mounting pressure at a

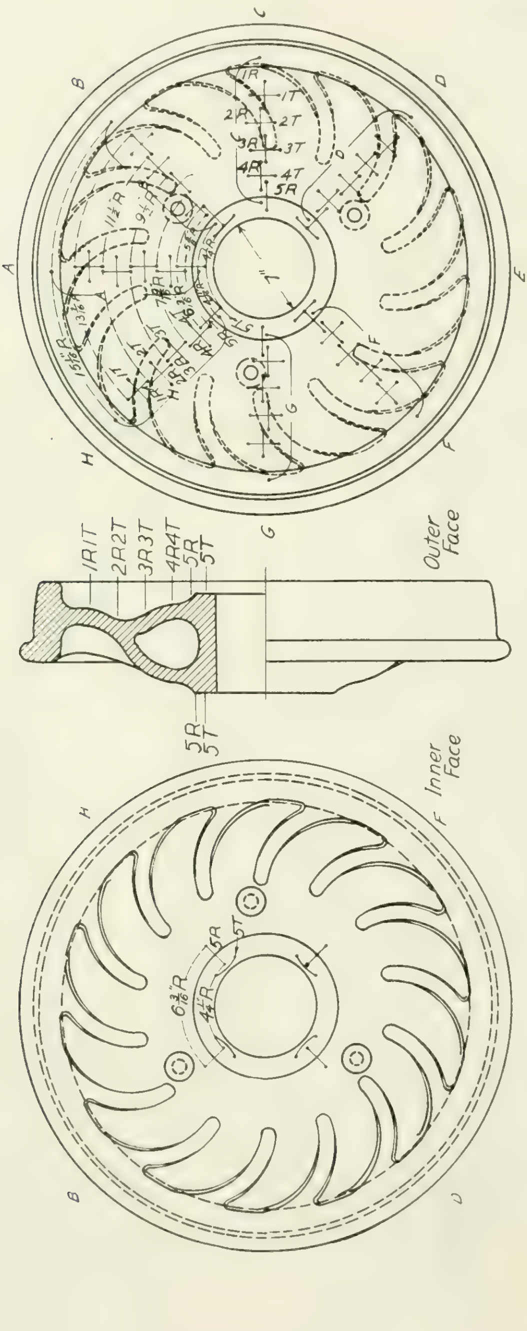

FIG. 10. LOCATION OF GAGE-LINES ON 33-IN. 725-LB. M. C. B. WHEEL No. 671 237

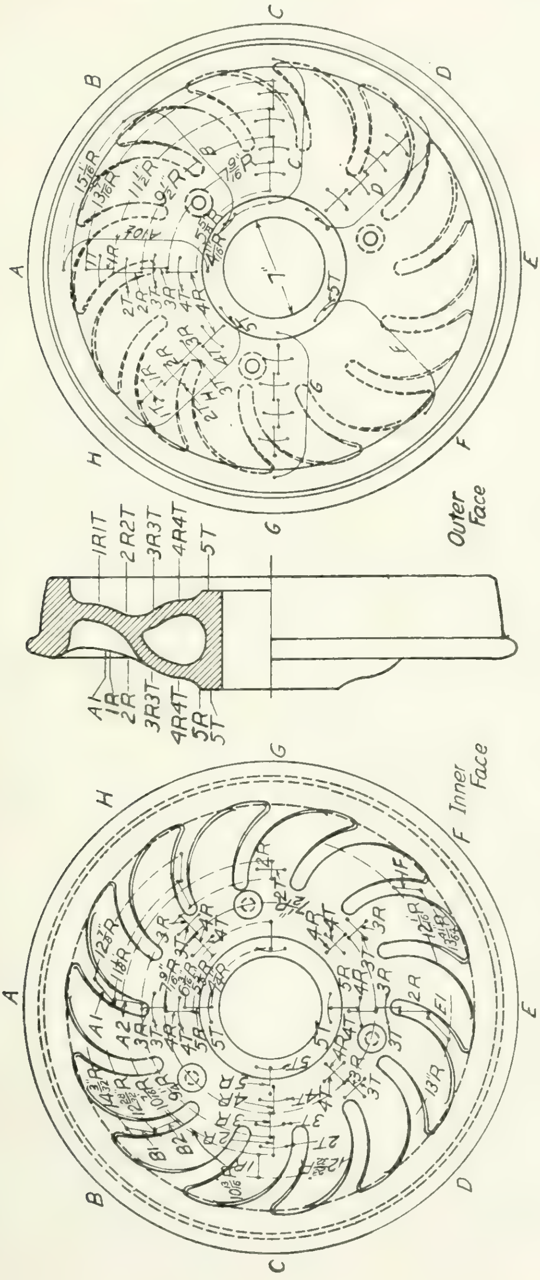

FIG. 11. LOCATION OF GAGE-LINES ON 33-IN. 725-LB. M. C. B. WHEEL No. 671 449

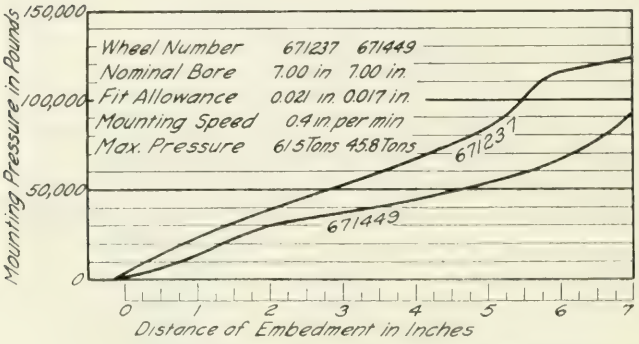

FIG. 12. AUTOGRAPHIC DIAGRAMS OF PRESSURES REQUIRED TO MOUNT TWO 33-IN. 725-LB. WHEELS ON AXLE

speed of 0.4 in. per minute was 61.5 tons; i.e., the allowance was 25 per cent greater than the manufacturers' maximum allowance, while the pressure required was 5 per cent less than might be recorded by commercial types of wheel presses, which, however, operate in some cases at much higher speeds. Likewise, in wheel No. 671 449 the fit allowance equalled the manufacturers' maximum, and in commercial work the expected pressure would be about 65 tons. The maximum recorded pressure for this wheel, however, was about 45.8 tons, or roughly equal to that which would occur with the minimum fit allowance of 0.011 in. A similar condition existed in later mounting tests. This apparently indicates that the pressure required to mount the wheel is a function of the mounting speed. It is also probable that the magnitude of the mounting pressure is dependent on the alignment of the wheel bore and axle during mounting. As no spherical blocks were used in these tests when mounting the wheel—nor are they used on wheel presses—it is possible that the maximum recorded mounting pressures are higher than would be the case if perfect alignment between bore and axle existed. Hence it would appear that fit allowance might be a better criterion for mounting wheels than pressure where the mounting speed falls considerably outside the range of speeds customarily used in wheel shops, and in cases where poor alignment may occur. Under no circumstances, however, would it be advisable to discontinue the use of the final mounting pressure as a check against the placing of improperly fitted wheels in service.

In the case of wheel No. 671 237 the maximum tensile unit strain, due to mounting (Fig. 13), was on gage-line *B4T* on the *B* radial of the outer face. The magnitude of this strain was 0.00222 which corresponds to a simple tensile stress of 17 400 lb. per sq. in. Inspection of Fig. 10 shows that the measurement was taken in close proximity to a chaplet, and this may partially account for the large value. On the inner face the same value is nearly reached on gage-line *H5T* where the strain was 0.00218, and the simple tensile stress corresponding thereto is 17 300 lb. per sq. in. With respect to the compressive strains the maximum was on the outer face on gage-line *F3R* and equalled 0.00127, which corresponds to a simple compressive stress of 17 600 lb. per sq. in. This is in the region near the intersection of the inner and outer plates. A relatively short radius

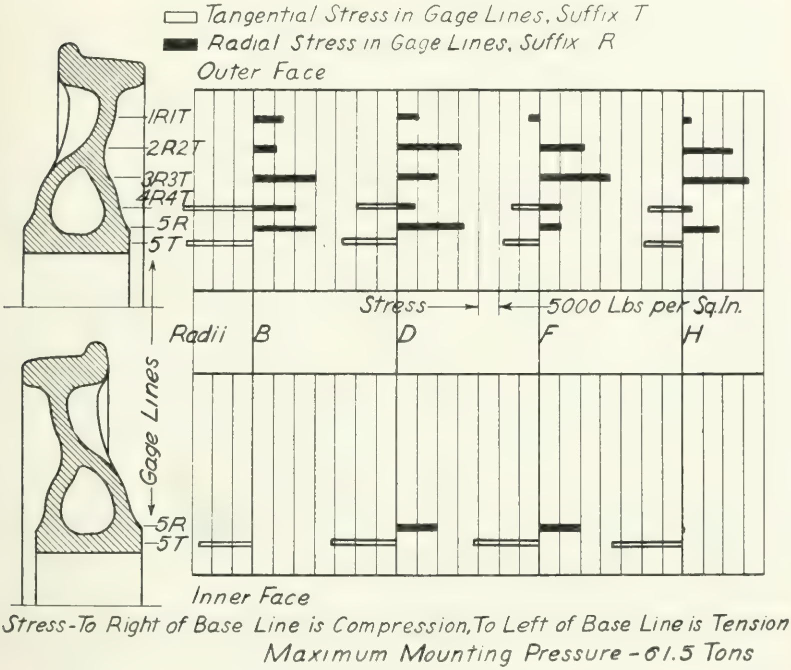

FIG. 13. CORRESPONDING SIMPLE UNIT STRESSES IN 33-IN. 725-LB. M. C. B. WHEEL NO. 671 237 CAUSED BY MOUNTING ON AXLE

of curvature, 3 in., exists on a portion of this gage-line, so that curved plate action may partially account for the values found in this region.

Fig. 14 presents the corresponding simple unit stresses produced in wheel No. 671 449 by mounting. A greater number of strain measurements were made on wheel No. 671 449, and a better idea of the magnitude of the stresses produced by forcing the wheel on the axle is possible than in the case of wheel No. 671 237. On the outer face the larger tangential or "hoop" stresses are in general near the bore, and they decrease toward the rim. This is readily apparent on radii *G* and *H,* and indicates a distribution of stresses which would be expected. There are, however, variations from this arrangement or distribution along radial lines *A, C,* and *D.* On these lines the tangential stress does not at all points show a decrease from bore to rim; at certain points the stresses are slightly greater than at other points nearer the bore. Bearing in mind, however, the fact that cast iron from the ordinary cupola cannot be cast so as to be of uniform strength, as was evidenced by the results of the previously mentioned tests made on specimens cut from various parts of the wheel, such variations could have been anticipated. Similar conditions occur on the inner face with respect to the tangential stresses. On this wheel the maximum recorded tensile strain occurred on the inner face in gage-line *A5T.* Its value was 0.00161 and the corresponding stress is 15 700 lb. per sq. in. A study of the radial strains, Fig. 14, on the outer face of wheel No. 671 449 reveals a distribution similar to that found in wheel No. 671 237. On all of the radial lines a tendency towards a concentration of stress appears on the outer face on gage-line *3R,* or near the intersection of the inner and outer plates. The maximum recorded strain occurred on gage-line *H3R,* and was equal to 0.00116, corresponding to 16 500 lb. per sq. in. On the inner face the stresses on both the radial and bracket gage-lines are in general compressive and of relatively small magnitude.

After being mounted, the wheels were subjected to static loads up to 200 000 lb. per wheel, or about 10 times the load they would be subjected to in practice. The maximum stresses in wheel No. 671 449, caused by combined mounting and static loads, are shown in Figs. 15 and 16. In Fig. 15 are shown the corresponding simple stresses caused by superimposing the effects of static loads, ranging from

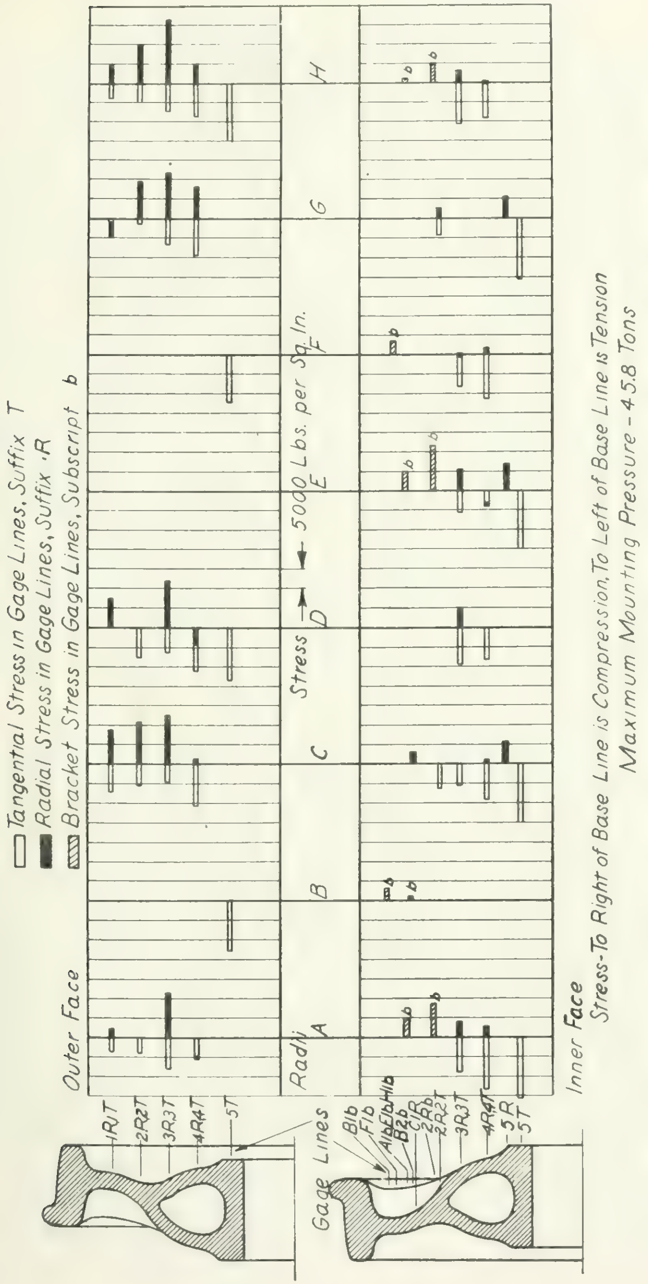

FIG. 14. CORRESPONDING SIMPLE UNIT STRESSES IN 33-IN. 725-LB. M. C. B. WHEEL NO. 671 449 CAUSED BY MOUNTING ON AXLE

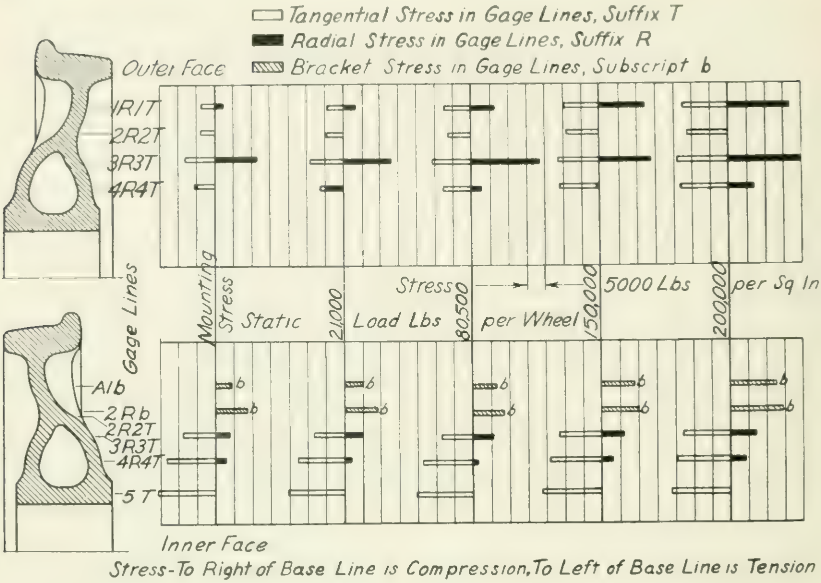

FIG. 15. CORRESPONDING SIMPLE UNIT STRESSES IN 33-IN. 725-LB. M. C. B. WHEEL NO. 671 449 DUE TO COMBINED EFFECTS OF MOUNTING AND STATIC LOADS. LOADS APPLIED ON *A* RADIAL

21 000 to 200 000 lb., upon those due to forcing the wheel on the axle. To prevent the axle being bent, it was given additional support between the wheels for wheel loads above 40 000 lb. In the tests represented by Fig. 15 the load was supplied on the *A* radial gage-line, i. e., the several loads were transmitted from the axle along this radial gage-line to the rail. The strains along this radial gage-line were larger than the strains along the other radial gage-lines; consequently, presentation of figures relating to the strains along the other radial gage-lines has been omitted from the discussion. In Appendix B, pages 81 to 85, Tables 2, 2a, 3, and 3a, there are presented in tabular form the numerical values of the strains measured on the various radial gage-lines of wheels No. 671 237 and 671 449. Fig. 16 presents for wheel No. 671 449 information similar to that in Fig. 15, except that for Fig. 16 the static loads were applied along the *G* radial gage-line.

With respect to the radial gage-line along which the load is applied, the stresses due to a static load, when imposed upon those

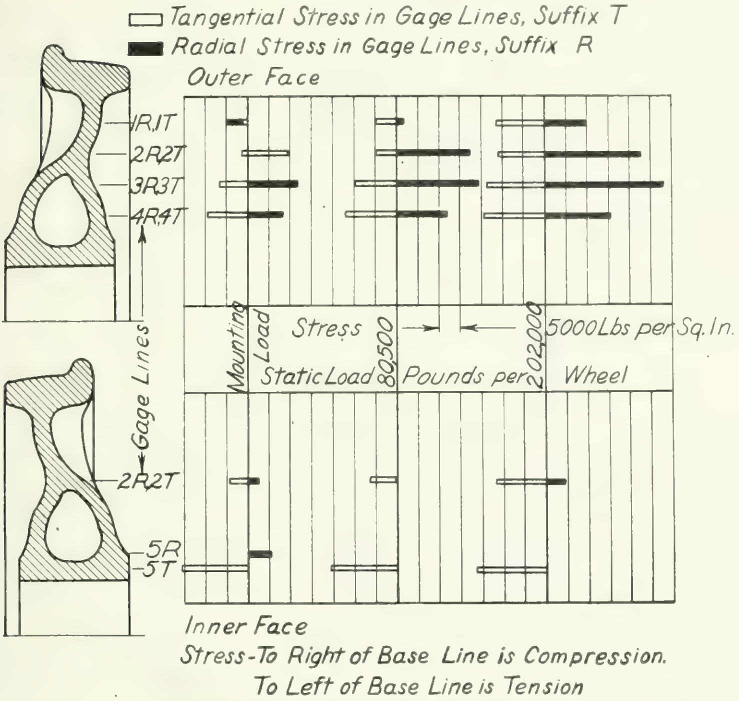

Fig. 16. Corresponding Simple Unit Stresses in 33-in. 725-lb. M. C. B. Wheel No. 671 449 Due to Combined Effects of Mounting and Static Loads. Loads Applied on *G* Radial

caused by mounting, are similar in kind to those of mounting, i. e., compressive on the radial and tensile on the tangential gage-lines. In ordinary practice the 33-in. 725-lb. wheel of this type may be subjected to a load not exceeding 20 125 lb. With a static load of 21 000 lb. the stresses are but slightly different from, and may be either greater or less than, those of mounting. This indicates that mounting is a much more important factor in producing stress than the maximum car load—neglecting impact and indirect effects—to which wheels are subjected in normal railroad service. With the static load equal to 200 000 lb. applied on the radial gage-line *A*, the maximum unit tensile strain due to the combined effects of mounting and load, was found to be on the tangential gage-line *A5T* on the inner face of the wheel, and to be equal to 0.00185. The corresponding unit stress is

16 500 lb. per sq. in., or an increase of only 800 lb. per sq. in. above that caused by mounting alone. The maximum increase in tensile strain occurred on the tangential gage-line *G1T* on the outer face when the 200 000 lb. load was applied along the *G* radial. For this loading, the increase in unit tensile stress on the several gage-lines over that due to mounting was as follows:

Gage-Line, Outer Face	Increase in Tensile Stress per sq. in.
G1T	10 500 lb.
G2T	9 100 lb.
G3T	6 900 lb.
G4T	4 700 lb.

As gage-line *G1T* was nearest the tread and *G4T* nearest the hub, these figures indicate that the influence of the static load decreases as the distance from the rail becomes greater. As a whole the observations relating to stresses produced by the static loads indicate that the influence of these loads (as expressed by increase of the tensile stresses) decreases as the distance from the rail toward the axle increases, although occasional exceptions to this rule occur.

Slightly different conditions exist on the radial gage-lines with respect to magnitude and distribution of the stresses. On the radial gage-lines the stresses are compressive, and the variations over the *G* radial gage-line for the corresponding 200 000-lb. load were:

Gage-Line, Outer Face	Increase in Compressive Stress per sq. in.
G1R	15 000 lb.
G2R	13 300 lb.
G3R	13 800 lb.
G4R	7 400 lb.

Here again the maximum increase in stress occurs nearest the tread and the minimum increase nearest the bore. It will be recalled that, in mounting, the opposite condition was found; namely, that the greatest stress occurred at the bore and the least at the tread. It is further evident from the above figures that the variation in stress in the radial direction differs from that in the tangential direction in that the latter shows an almost uniform decrease in the intensity when traversing the section from *G1T* to *G4T*, while in the former no such uniform decrease occurs. For the combined effect of mounting and the 200 000-lb. static load the maximum recorded compressive strain was on radial gage-line *G3R*, and amounted to 0.00267. The corresponding simple compressive stress is 25 600 lb. per sq. in., or 13 800 lb. greater than the stress produced by mounting alone. The results on which Figs. 13 to 16 inclusive are based, together with the measured strains, are given in Appendix B, Tables 2, 2a, 3, and 3a. Figures simliar to Figs. 15 and 16, which relate to wheel No. 671 449, are not submitted for wheel No. 671 237, due to the fact that the measurements of strain taken during the mounting test of wheel No. 671 237 were so meagre; the numerical values of these, however, are given in Appendix B, Table 2a.

In summarizing, these tests would indicate that, directly, the wheel load of ordinary service does not materially alter the existent strains caused by mounting, although, indirectly, the wheel load is a factor in producing stress, due to the bearing it has on flange pressure, impact, etc. Load application results in increasing the compressive strains already existent in the wheel to a greater extent than it increases the tensile strains. It is further evident that in the absence of speed and track curvature, very heavy wheel loads may be sustained without greatly increasing the magnitude of the tensile stresses which were produced by mounting.

14. *Corresponding Simple Stresses in Two 33-in. 740-lb. Arch Plate Wheels Due to Mounting Wheels on Axle, together with Combined Effects of Mounting and Static Loads.*—From the results of the tests on the wheels mentioned in the preceding section it was impossible to determine the exact position of either the maximum tensile or the maximum compressive stress. Furthermore, the relatively small number of strain lines prevented a clear conception of the stress distribution across the wheel section. Accordingly, in subjecting a pair of 33-in. 740-lb. Arch Plate wheels to the mounting and static load tests, they were prepared with approximately twice as many gage-lines as were used on the previously tested wheels. This was done by placing the 2-in. gage-lines 1 in. apart, instead of 2 in. apart as in the former tests. The locations of the gage-lines on wheels Nos. 04 474 and 04 476 are given in Figs. 17 and 18 respectively. The system used for identification of the gage-lines is shown in the figures,

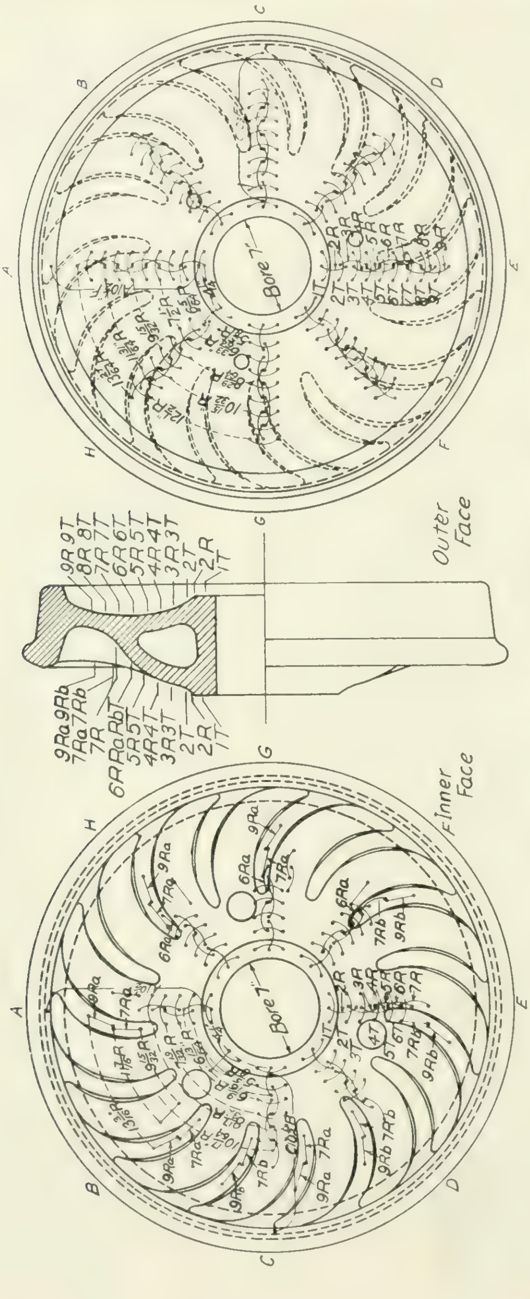

FIG. 17. LOCATION OF GAGE-LINES ON 33-IN. 740-LB ARCH PLATE WHEEL NO. 04 474

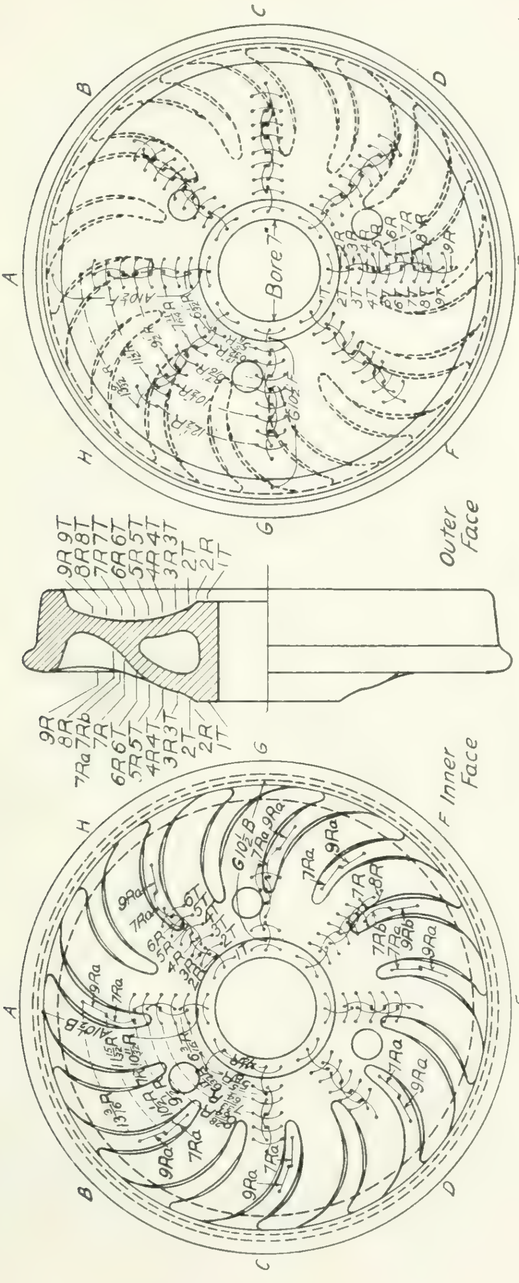

FIG. 18. LOCATION OF GAGE-LINES ON 33-IN. 740-LB ARCH PLATE WHEEL No. 01476

and is similar to that used on the 33-in. 725-lb. wheels* except that the gage-lines are numbered from the hub outward to the tread.

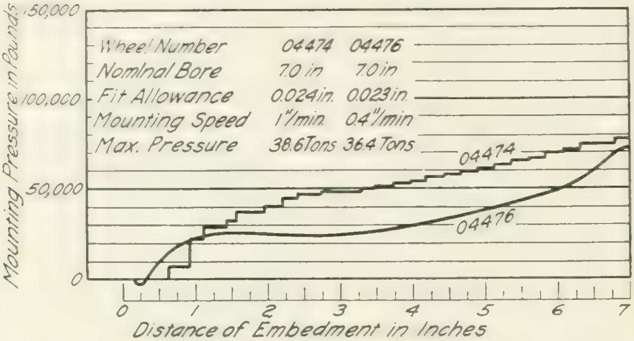

FIG. 19. AUTOGRAPHIC DIAGRAMS OF PRESSURES REQUIRED TO MOUNT TWO 33-IN. 740-LB. ARCH PLATE WHEELS ON AXLE

In Fig. 19 are given the fit allowance, the mounting speed, and the pressure required to force the axle into the wheels. It will be noticed that the mounting speed of wheel No. 04 474 is 2.5 times that of wheel No. 04 476. The results on the 725-lb. M. C. B. wheels suggested that the speed of mounting might be a factor in the magnitude of the final pressure. Although wheel No. 04 474 had the higher pressing speed, yet its higher recorded pressure can hardly be ascribed to this cause, as the slightly greater final pressure, 4.2 tons, may have been due to other factors, such as fit allowance, character of wheel material, etc. A series of tests with a greater variation in speed would probably be necessary to satisfactorily answer this question.

The corresponding simple stresses caused by forcing the wheels on the axle are shown in Figs. 20 and 21. The values of the stresses are based on the measured strains and the assumed stress-strain relation given by Fig. 6. If the stress at some particular radius is taken, it will be noticed in Figs. 20 and 21 that there may be a considerable variation between the maximum and the minimum values occurring on the several radial gage-lines *A* to *H*. As a result of this, the variations in the intensities of the stresses on one radial gage-line may be entirely different from the variations on another, although adjacent. To obtain a diagram of the stress distribution across the section of the wheel, more representative of the wheel as a whole and better for

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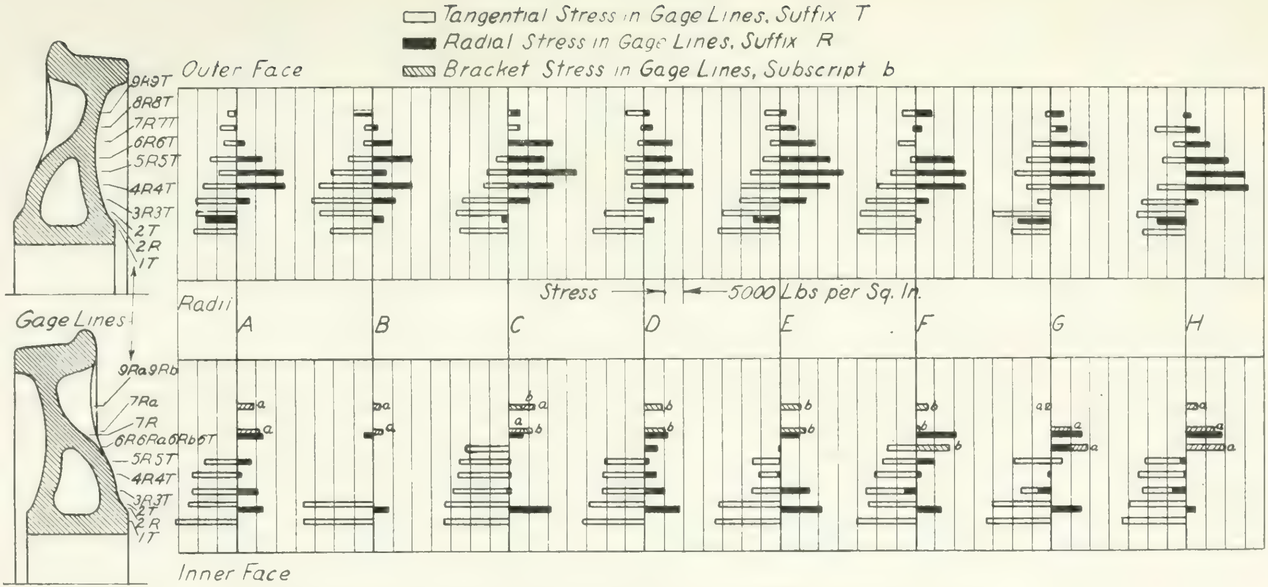
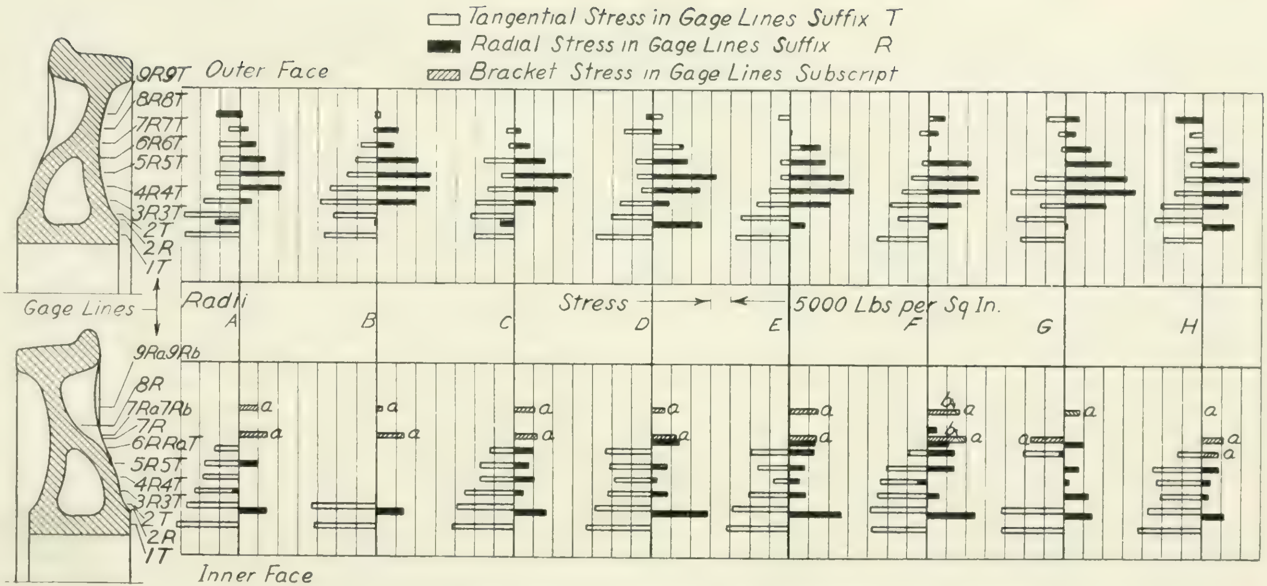


Stress-To Right of Base Line is Compression, To Left of Base Line is Tension
Maximum Mounting Pressure - 38.6 Tons

FIG. 20. CORRESPONDING SIMPLE UNIT STRESSES IN 33-IN. 740-LB. ARCH PLATE WHEEL NO. 01 474 CAUSED BY MOUNTING ON AXLE



Stress-To Right of Base Line is Compression, To Left of Base Line is Tension

Maximum Mounting Pressure 36.4 Tons

Fig. 21. Corresponding Simple Unit Stresses in 33-in. 740-lb. Arch Plate Wheel No. 04 476 Caused by Mounting on Axle

purposes of comparison, the measured strains on the several radii at corresponding distances from the center of the wheel were averaged. The corresponding simple stresses for these average strains were then determined and plotted, as shown in Figs. 22 and 23. This method of determining the average stresses gives somewhat higher values than would be obtained by averaging the stresses corresponding to the measured strains.

Inspection of Figs. 22 and 23 immediately suggests the stress distribution across the section. The stresses on the tangential gage lines on both the inner and outer faces are tensile, largest near the bore and decreasing towards the tread. No such simple arrangement, however, exists in the stresses produced on the radial gage-lines. On these lines on the outer face when traversing the section from bore to tread a relatively small stress, either compression or tension, exists on gage-line *2R*, and the character of the stress then becomes compressive, reaching a maximum on either *4R* or *5R*—just previous to the joining of the inner and outer plates—after which the compressive stress again decreases. On the radial gage-lines *2R* on the outer face of wheel No. 04 474, Fig. 20, the strain is an elongation, contrary to what might be expected. The reason for this condition is not apparent but it is perhaps due to a bending action at that point producing tension of a magnitude sufficient to change the compressive stress due to mounting into a tensile stress. On the outer face of both wheels the variation in the radial stresses in the regions represented by gage-lines *2R* to *5R* inclusive suggests a condition of bending combined with compression due to mounting.

On the inner face of wheel No. 04 474 the effect of bending is evident on the radial gage-lines. The stress on these lines is compressive and reaches a maximum at the bore. As the radial distance from the center increases, the stress decreases until a minimum is reached near *4R*, after which it increases up to *7R*. Beyond *7R* no readings were obtained because of the interference of the wheel brackets with the strain gages, but it is reasonable to assume that a decrease in stress will develop soon after *7R* is passed. The comparatively short radii of curvature in this portion of the wheel without doubt greatly affect any bending action which takes place and are largely responsible for the variation occurring in the magnitude of

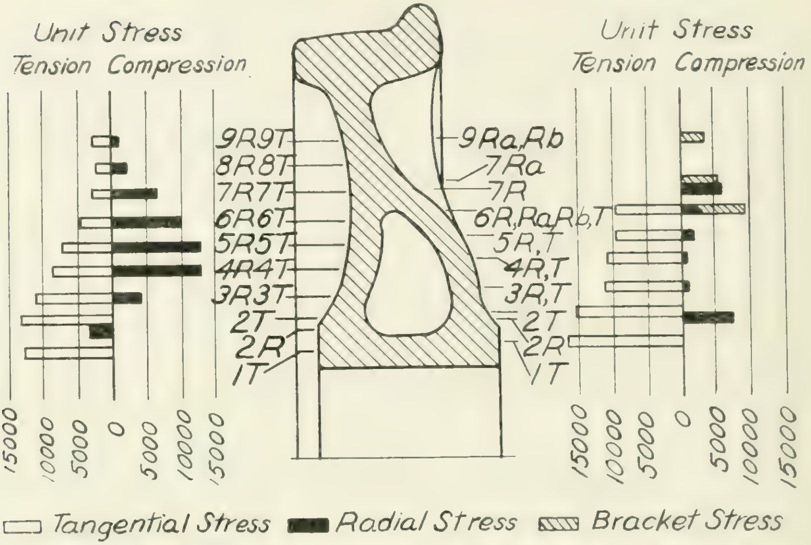


FIG. 22. AVERAGE UNIT STRESSES IN 33-IN. 740 LB. ARCH PLATE WHEEL NO. 04 474 CAUSED BY MOUNTING ON AXLE

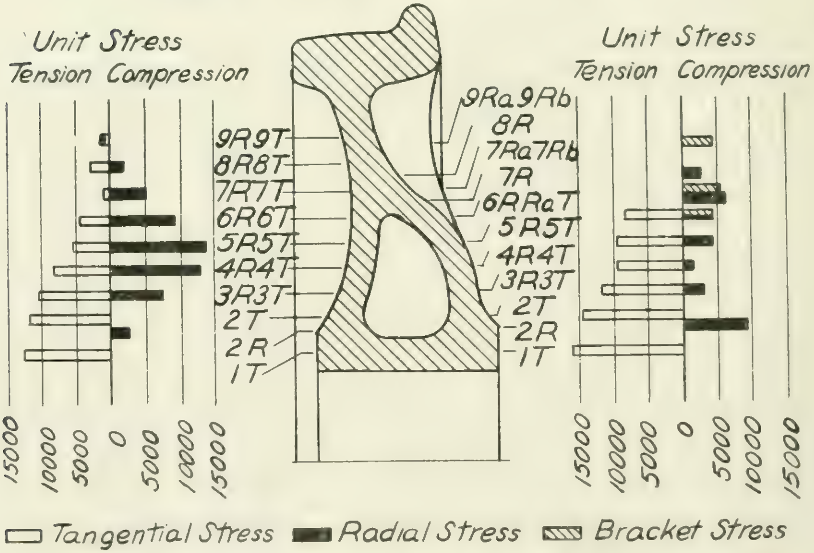


FIG. 23. AVERAGE UNIT STRESSES IN 33-IN. 740-LB. ARCH PLATE WHEEL NO. 04 476 CAUSED BY MOUNTING ON AXLE

the stresses. Similar conclusions may be reached with respect to the stresses on the radial gage-lines on the inner face of wheel No. 04 474 as shown in Fig. 22.

In wheel No. 04 474 the maximum average tensile and compressive strains are 0.00194 and 0.00084, located on tangential gage-lines *1T* of the inner face, and radial gage-lines *5R* of the outer face, respectively. The corresponding tensile stress is 16 800 lb. per sq. in. and the corresponding compressive stress 13 000 lb. per. sq. in. In the other wheel, No. 04 476, the positions of the maximum average tension and compression are identical with those of wheel No. 04 474. The tensile stress, however, is 16 000 lb. per sq. in., i. e., slightly lower than in wheel No. 04 474, while the compressive stress is 13 700 lb. per sq. in., or 700 lb. higher than in wheel No. 04 474. The fit allowance on these two wheels differs by only 0.001 in. and a close agreement exists both as to magnitude and location of the maximum strains or stresses caused by mounting.

These wheels were also subjected to loads equalling 200 000 lb. per wheel, and the combined effects of mounting and the various static loads are shown in Figs. 24 to 27, inclusive. The corresponding numerical values are tabulated in Appendix B, Tables 4, 4a, 5, and 5a.

From the diagrams showing the combined effects of mounting and static loads, Figs. 24 to 27, inclusive, it is evident that the metal in a radial direction on the outer face of the wheel is in general in compression. An exception to this occurs on gage-line *2R*, where tension is indicated under several loads. Here again the tension is probably due to flexure in conjunction with the direct thrust of mounting. It will be noticed in addition that the intensity of the compressive stress on the outer face exceeds that on the inner face, this fact indicating that the static load is transmitted from rail to hub mainly through the outer plate, while the smaller portion of the load is carried through the inner plate. In general the magnitude of the tensile stress is slightly greater on the inner plate of the wheel than on the outer plate. In the curved portion bending assists in the production of tensile stress. The maximum compressive stress due to combined effects of mounting and static load may occur on either of the radial gage-lines *4R* or *5R* on the outer face, i.e., at a point in the outer plate just before it is joined with the inner plate. The

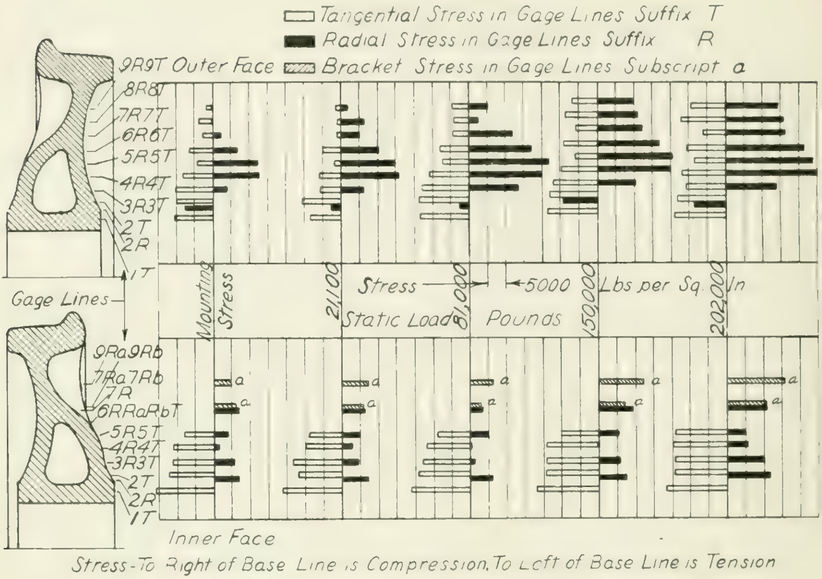


Fig. 24. Corresponding Simple Unit Stresses in 33-in. 740 lb. Arch Plate Wheel No. 04 474 Due to Combined Effects of Mounting and Static Loads. Loads Applied on A Radial

maximum tensile stress is found on either of the tangential gage-lines *1T* or *2T* on the inner face, that is, in the region of the hub. The conclusions to be drawn from the tests on these wheels are similar to those reached with regard to the 33-in. 725-lb. M. C. B. wheels—namely, that under load

(1) the strain or stress due to static load is similar in kind to that of mounting, i. e., compressive on the radial gage-lines and tensile on the tangential gage-lines;

(2) static load increases the magnitude of the compressive stresses on the radial gage-lines to a greater extent than it increases the magnitude of the tensile stresses on the tangential gage-lines;

(3) for the normal service load the maximum strains or stresses are not greatly different from the maximum strains or stresses due to mounting;

(4) abnormally heavy wheel loads may be sustained in the absence of such other factors as side thrust, impact, etc., without greatly increasing the intensity of the tensile stresses over values already existent through pressing the wheel on the axle.

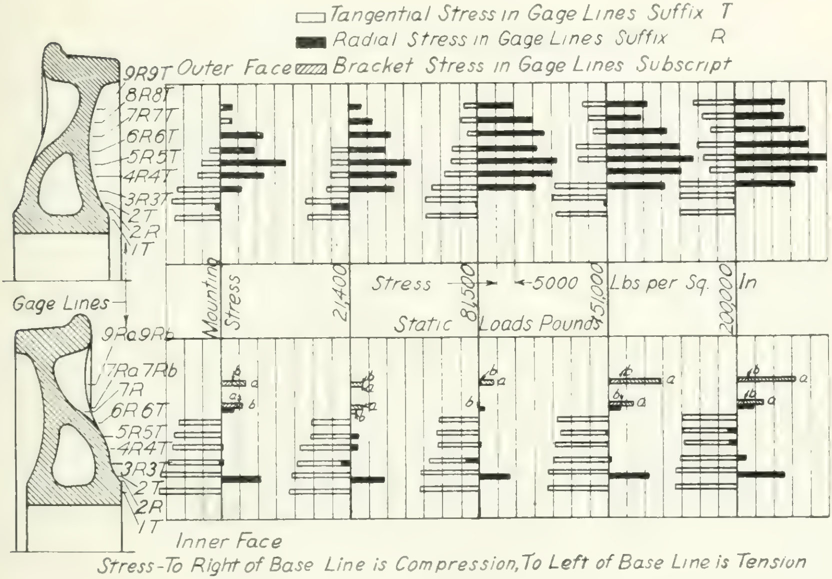
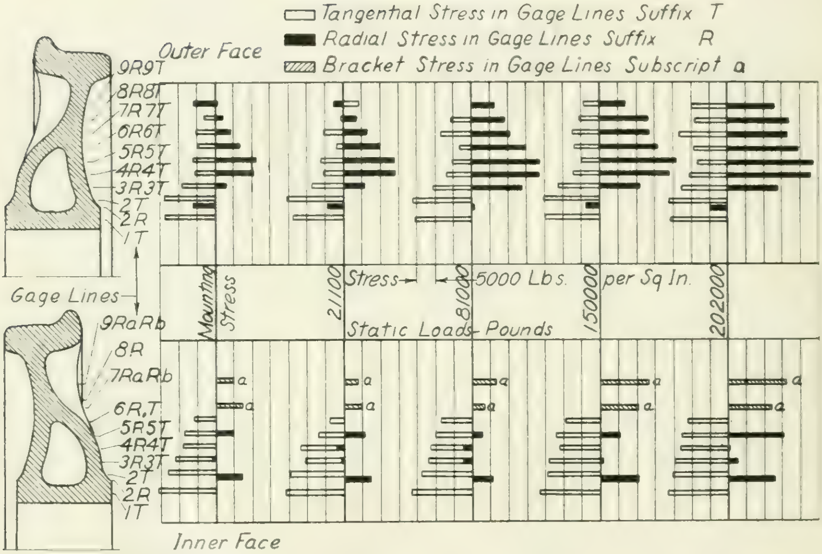


Fig. 25. Corresponding Simple Unit Stresses in 33-in. 740-lb. Arch Plate Wheel No. 04 474 Due to Combined Effects of Mounting and Static Loads. Loads Applied on *C* Radial

It would have been desirable to examine the relative ability of the two types of wheels, as represented by the 725-lb. M. C. B. wheel and the 740-lb. Arch Plate wheel, to withstand the stresses produced by forcing the wheel on the axle and by the application of load. Due to their different fit allowances, however, and also to the relatively few strain measurements made on the 725-lb. wheels, no satisfactory comparison along these lines has been found possible.



Stress-To Right of Base Line is Compression, To Left of Base Line is Tension

FIG. 26. CORRESPONDING SIMPLE UNIT STRESSES IN 33-IN. 740-LB. ARCH PLATE WHEEL NO. 04 476 DUE TO COMBINED EFFECTS OF MOUNTING AND STATIC LOADS. LOADS APPLIED ON *A* RADIAL

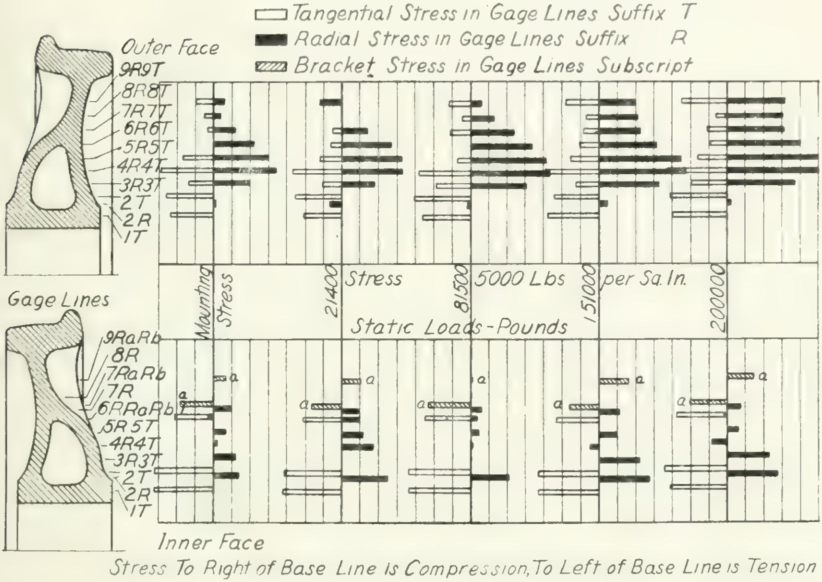


FIG. 27. CORRESPONDING UNIT STRESSES IN 33-IN. 740-LB. ARCH PLATE WHEEL NO. 04 476 DUE TO COMBINED EFFECTS OF MOUNTING AND STATIC LOADS. LOADS APPLIED ON *G* RADIAL

VI. RESULTS FROM ADDITIONAL MOUNTING TESTS

15. *Preparation of Wheels for Mounting.*—In order to determine in greater detail the effect of mounting, two 33-in. 625-lb. M. C. B., and two 33-in. 725-lb. M. C. B. wheels were subjected to this type of test. In forcing a wheel on an axle it is reasonable to expect higher stresses near the bore than at points distant from the bore. In the preceding tests it was impossible to take measurements at points less than 3/4 in. from the bore. This condition arose through interference of the axle with the strain gage. To overcome this interference and thereby allow measurments to be taken in close proximity to the bore, the wheels were mounted on stub, or short, axles whose length equalled the thickness of the wheel at the bore. The location of the gage-lines on the several wheels is shown in Figs. 28 to 31, inclusive. The additional gage-lines in the hub region will be noticed. By the use of these stub axles it was possible to get measurements relating to tangential stresses within 3/16 in. of the bore, and from these to get an estimate of the intensity of tangential or "hoop" stresses at the bore. To determine the intensity of stress in the region of the core holes on the inner face of the wheels, a few gage-lines were placed near certain core holes. These gage-lines are shown on the various figures. For a similar reason one of the wheels, No. 49 317, had gage-lines on the outer face in the vicinity of the chaplets.

In the case of the previously tested wheels the fit allowances were so nearly alike in each of the pairs of wheels tested that nothing as to the effect of fit allowance for a certain type of wheel could be determined. Accordingly, for the tests about to be described, certain fit allowances were chosen so that information might be obtained with regard to the effect of this factor.

16. *Corresponding Simple Stresses Due to Forcing Two 33-in. 625-lb. M. C. B. Wheels, with Different Fit Allowances, on Axles.*—The autographic diagrams taken while mounting these wheels are given in Fig. 32. Wheel No. 809 711 had a fit allowance of 0.009 in. and the maximum pressure was 14.4 tons. For this fit, the wheel was bored smooth with a tool of large radius, feed being approximately 1/27 in. per revolution. The allowance in mounting wheel No. 822 269

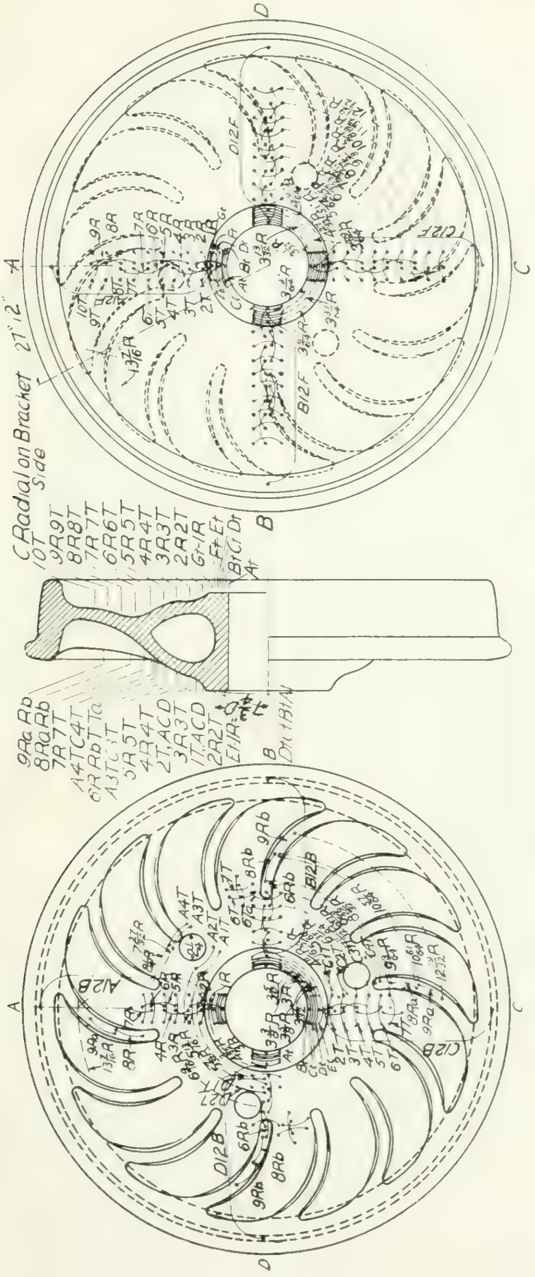


FIG. 28. LOCATION OF GAGE-LINES ON 33 IN. 625-LB. M. C. B. WHEEL NO. 809 711

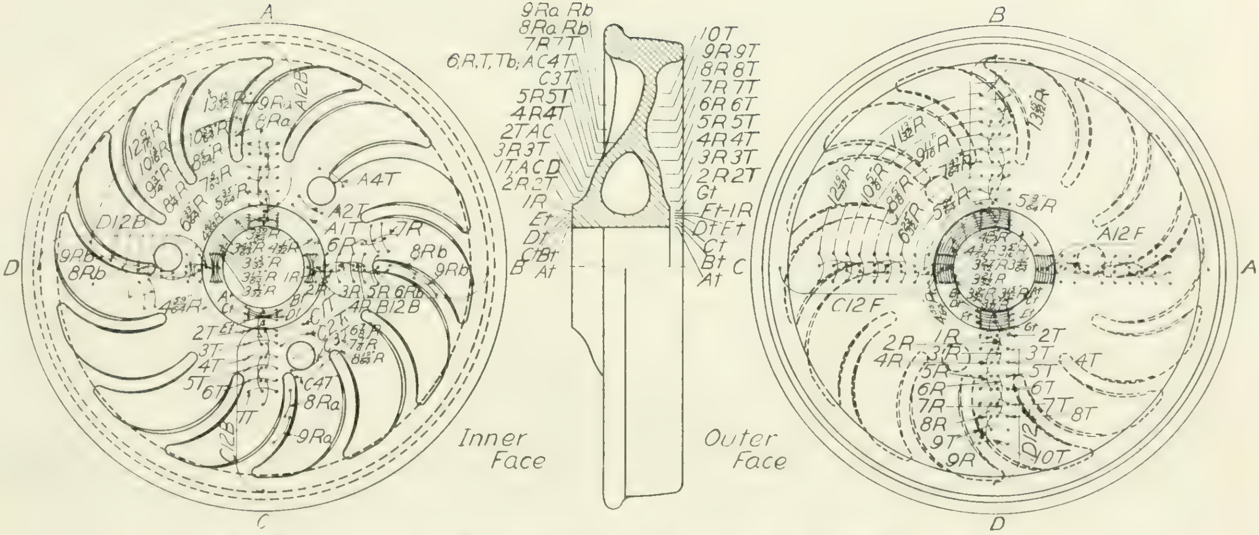


Fig. 29. Location of Gage-Lines on 33-in. 625-lb. M. C. B. Wheel No. 822 269

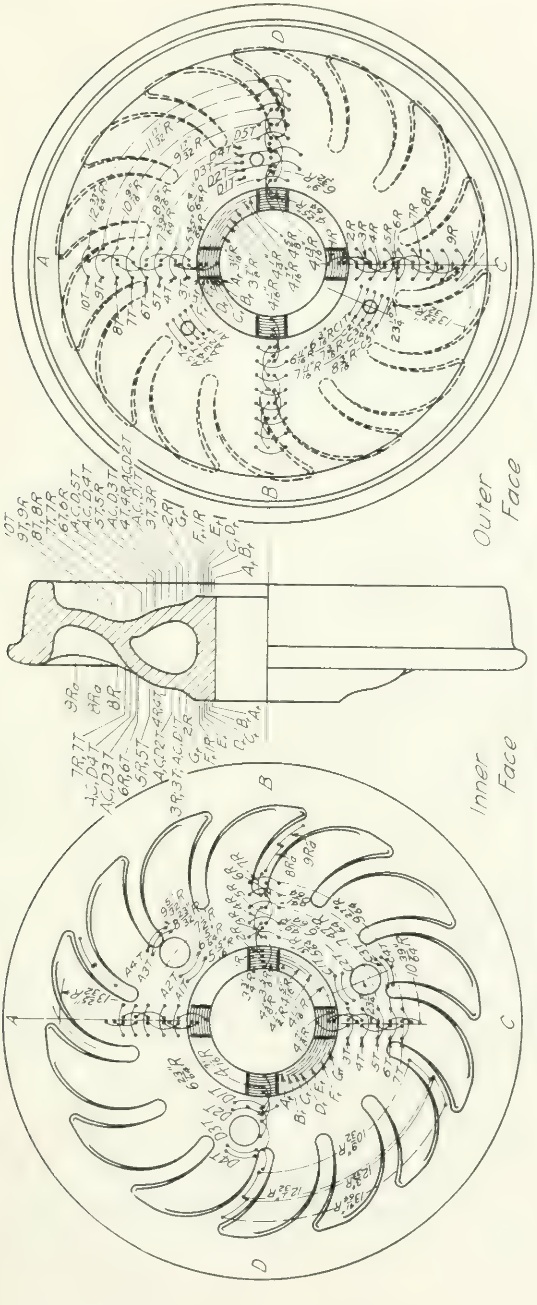


Fig. 30. Location of Gage-Lines on 33-in. 725-lb. M. C. B. Wheel No. 49 317

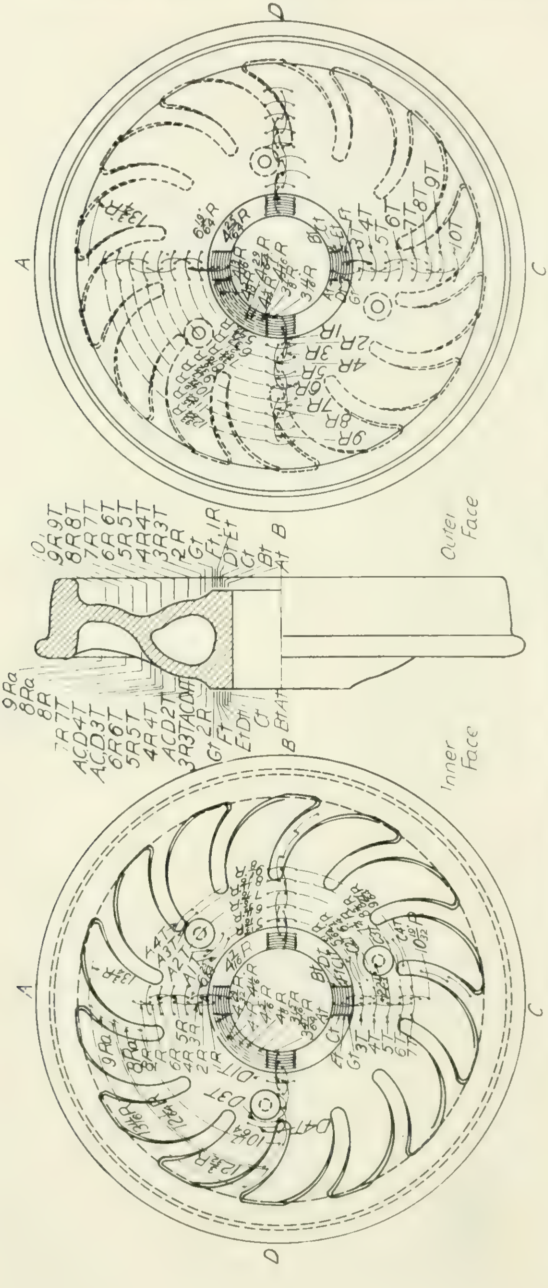


FIG. 31. LOCATION OF GAGE LINES ON 33-IN. 725-LB. M. C. B. WHEEL No. 49 305

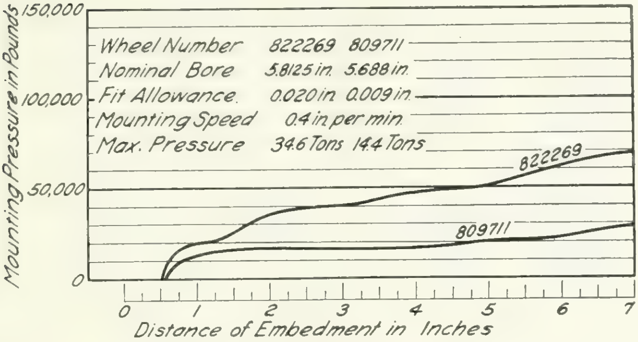
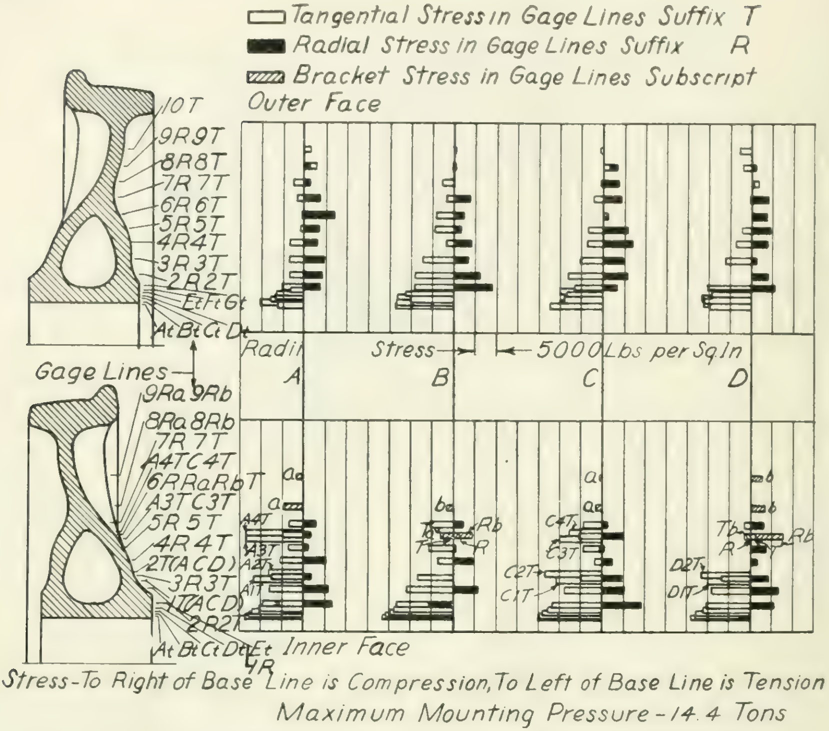


FIG. 32. AUTOGRAPHIC DIAGRAMS OF PRESSURES REQUIRED TO MOUNT TWO 33-IN. 625-LB. M. C. B. WHEELS ON AXLE

was 0.020 in., and the final pressure recorded was 34.6 tons. This wheel was also bored smooth and the axle showed no evidence of filing, the tool marks being plainly visible and indicating a feed of 1/14 in. per revolution. The rate of mounting was 0.4 in. per min. in both cases.

The stresses set up on the various gage-lines of these two wheels are given in Figs. 33 and 34. The numerical results on which these figures are based are given in Appendix B, Tables 6 and 7. Here again variation exists in the intensity of the stress at uniform distances from the center, but on different radial lines. The stresses in wheel No. 822 269 are higher than those of wheel No. 809 711, a condition which would be expected, due to the fact that the former wheel had the larger fit allowance. In these two wheels the measurements were made on four radial lines. The four series of strain readings taken on the four radial lines were averaged and the corresponding stresses plotted in Figs. 35 and 36 in order to obtain a more representative diagram showing the variation of stress from point to point, and the effect of the different fit allowances. A comparison of these figures plainly shows the larger stresses in wheel No. 822 269 caused by the larger fit allowance. The trend of the variation in stress from bore to tread is not as clearly evident in Fig. 35 as in Fig. 36 on account of the relatively small values of the stresses shown in Fig. 35. Inspection of Fig. 36, however, suggests substantially the same conclusions as



Stress-To Right of Base Line is Compression, To Left of Base Line is Tension
Maximum Mounting Pressure - 14.4 Tons

Fig. 33. Corresponding Simple Unit Stresses in 33-in. 625-lb. M. C. B. Wheel No. 809 711 Caused by Mounting on Axle

were drawn in connection with the mounting tests previously discussed.

(1) The tangential or hoop stress on the outer face is tension and reaches a maximum at the bore; this stress decreases in intensity as the radial distance from the bore increases.

(2) In a radial direction on the outer face the stress is compression and similarly is a maximum at the bore, and, as the radial distance from the bore increases, its magnitude at first decreases, then increases up to a point near the intersection of the inner and outer plates, beyond which a decrease again occurs.

(3) The tangential stress on the inner face is tension and varies in a manner similar to that on the outer face, except that the decrease is not as uniform.

(4) In the radial direction on the inner face the stress is compressive, reaching a maximum at the bore, and as the radial distance from the bore increases the stress decreases until near the intersection of the inner and outer plates, where a tendency towards an increase, or an actual increase, occurs; beyond this region the stress again decreases.

(5) The tensile stresses on the tangential gage-lines adjacent to the core holes on the inner plate are almost equal in value to those on the tangential gage-lines nearest the bore. The stresses in the brackets caused by mounting are relatively insignificant.

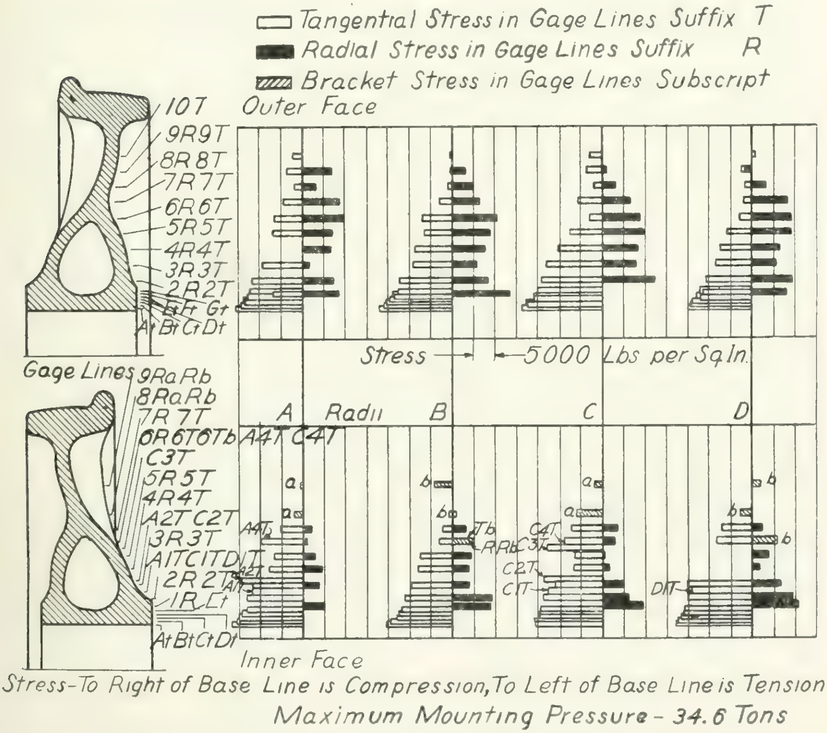


FIG. 34. CORRESPONDING SIMPLE UNIT STRESSES IN 33-IN. 625-LB. M. C. B. WHEEL NO. 822 269 CAUSED BY MOUNTING ON AXLE

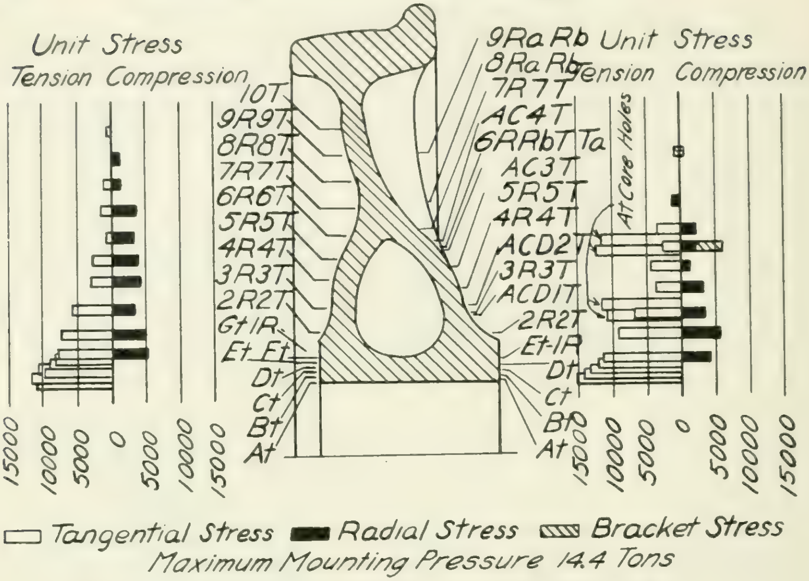


FIG. 35. AVERAGE UNIT STRESSES IN 33-IN. 625-LB. M. C. B. WHEEL NO. 809 711 CAUSED BY MOUNTING ON AXLE

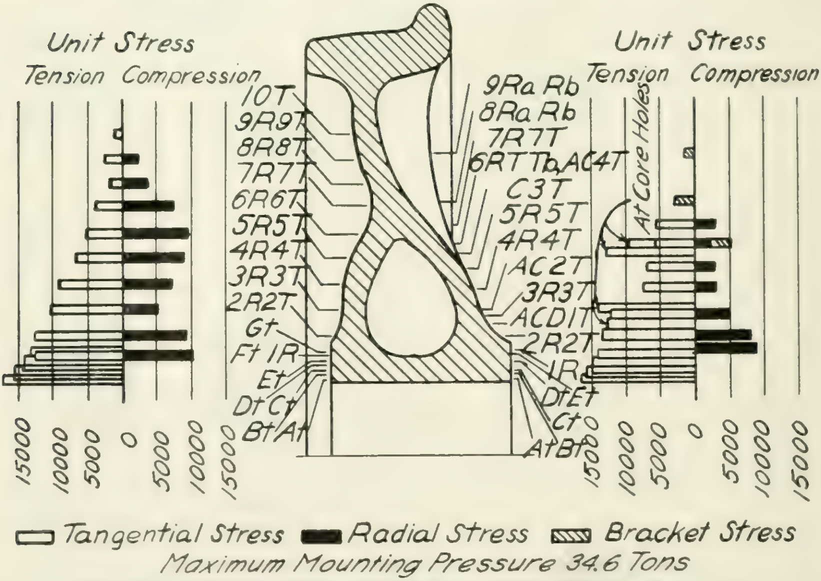


FIG. 36. AVERAGE UNIT STRESSES IN 33-IN. 625-LB. M. C. B. WHEEL NO. 822 269 CAUSED BY MOUNTING ON AXLE

17. *Corresponding Simple Stresses Due to Forcing Two 33-in. 725-lb. M. C. B. Wheels, with Different Fit Allowances, on Axles.*—For this test, wheels Nos. 49 317 and 49 305 had fit allowances of 0.0095 and 0.0136 in. respectively, while the corresponding final pressures were 39.9 and 43.0 tons. The rate of pressing was 0.4 in. per min. in each case. The mounting diagrams are shown in Fig. 37. As in the case of the 625-lb. wheels, one axle was filed smooth and the other retained the tool marks. As far as the stresses produced by mounting are concerned, there was nothing in these tests that would indicate the relative effects of a smooth or a rough axle. In Figs. 38 and 39 are shown the stresses corresponding to the measured strains caused by mounting. The corresponding results are presented in Appendix B, Tables 8 and 9. Appendix B also presents results for the average effects which are shown in Figs. 40 and 41. Although differences exist in the relative ability of the 625 and 725-lb. wheels to withstand the stresses produced by mounting, yet there is a marked similarity in the location of the maximum and minimum stresses. The conclusions to be drawn with reference to the 725-lb. wheel are in general identical with those for the 625-lb. type as given in Section 16, page 58 of this bulletin. In addition, Fig. 40 shows that the intensity of the tensile stress on the tangential gage-lines in the regions of the chaplets on the outer face nearly approaches that occurring at the bore.

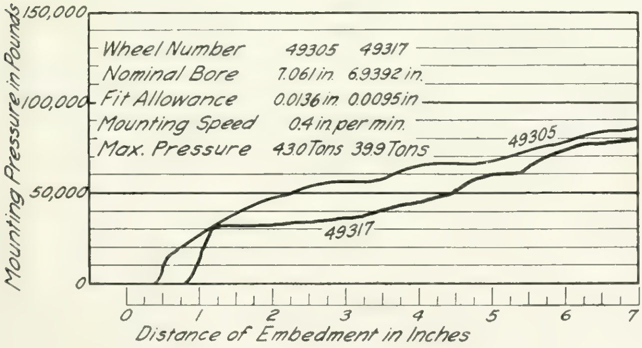


FIG. 37. AUTOGRAPHIC DIAGRAMS OF PRESSURES REQUIRED TO MOUNT TWO 33-IN. 725-LB. M. C. B. WHEELS ON AXLE

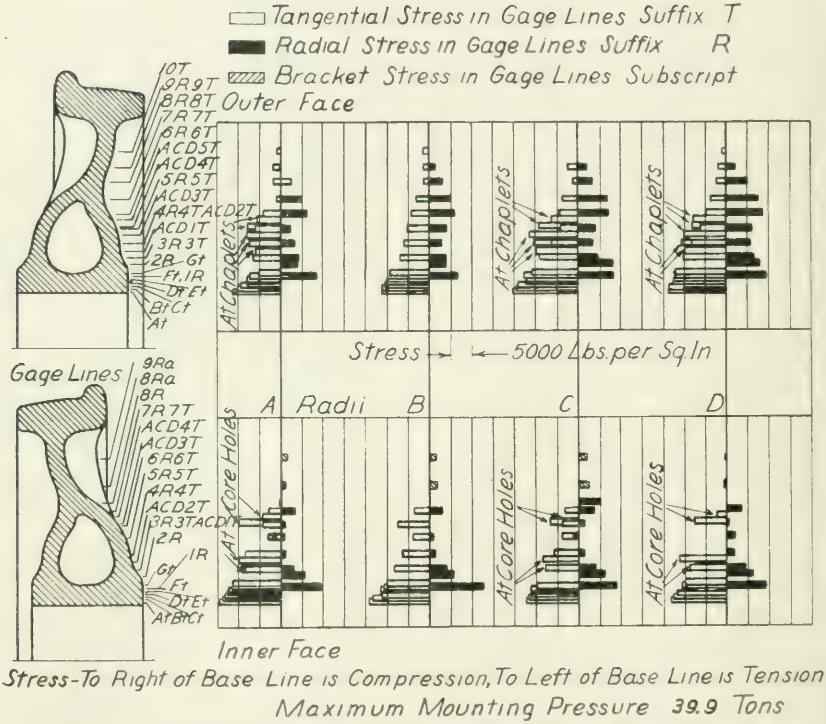


FIG. 38. CORRESPONDING SIMPLE UNIT STRESSES IN 33-IN 725-LB. M. C. B. WHEEL NO. 49 317 CAUSED BY MOUNTING ON AXLE

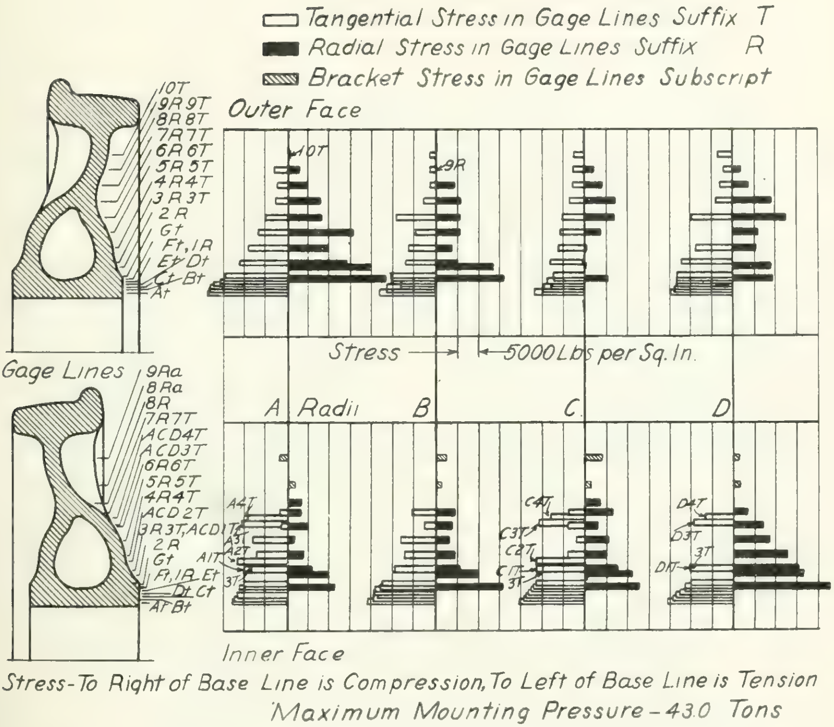


Fig. 39. Corresponding Simple Unit Stresses in 33-in 725-lb. M. C. B. Wheel No. 49 305 Caused by Mounting on Axle

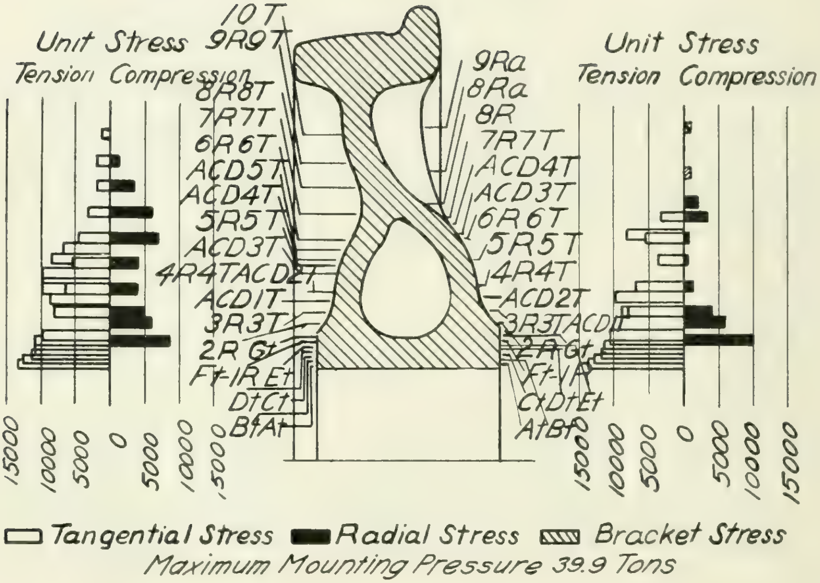


FIG. 40. AVERAGE UNIT STRESSES IN 33-IN. 725-LB. M. C. B. WHEEL NO. 49 317 CAUSED BY MOUNTING ON AXLE

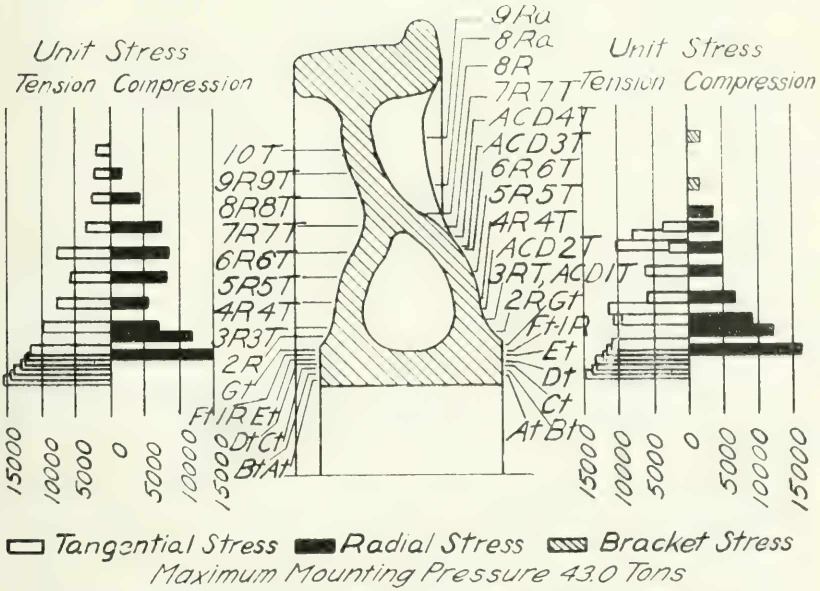


FIG. 41. AVERAGE UNIT STRESSES IN 33-IN. 725-LB. M. C. B. WHEEL NO. 49 305 CAUSED BY MOUNTING ON AXLE

VII. Summary of Conclusions

The results recorded in the previous pages may be summarized as follows:

(1) The tensile strength of the metal taken from different parts of the plates of three wheels ranged from 23 300 to 32 800 lb. per sq. in., and the modulus of elasticity ranged from 14 to 28 million lb. per sq. in. It is probable that these variations may be explained by variations in chemical composition of the several specimens, and variations in the treatment of the wheels after casting. Whatever be the reason, however, they suggest that a study of the metallurgy of wheel irons offers possibilities of improvement by which the higher values obtained in these tests might be consistently maintained or possibly exceeded.

(2) No distinct relation was apparent between the ultimate strength of wheel iron and either the Brinell or the scleroscope hardness, nor could a constant relation be determined between the Brinell and scleroscope results.

(3) In forcing the 625-lb. and 725-lb. M. C. B. or Washburn type of wheel on an axle the maximum tensile strain or stress is a tangential or ''hoop'' strain or stress occurring at the bore, and it may be on either the inner or the outer face of the wheel.

On the outer face of this type of wheel in a radial direction the strains or stresses are compressive. They are a maximum at the bore and, in traversing the section of the wheel from the bore toward the tread, they decrease up to a point where the radius equals the mean radius of the core, beyond which an increase occurs up to a point at or near the intersection of the inner and outer plates, after which a decrease again occurs.

On the inner face in a radial direction the strains or stresses are likewise compressive and a maximum at the bore. If a similar traverse be made across the section, these strains or stresses decrease up to a point where the radius equals the mean radius of the core; they are then of approximately uniform intensity up to a point whose radius is equal to the outer radius of the core; beyond this point they again decrease.

(4) In pressing the 740-lb. Arch Plate wheels on the axle, the maximum tensile strain or stress was a tangential strain or

stress on the inner face and at the bore. In general the tensile stresses on either face of the wheel were a maximum at the bore and decreased toward the tread.

With respect to the strains or stresses in a radial direction on the outer face, no measurements were taken in close proximity to the bore, and accordingly nothing definite can be stated concerning their intensity in this region. However, the strain in a radial direction taken nearest to the bore was relatively small and was tension in one wheel and compression in the other. This fact may be due to bending action in conjunction with the thrust of mounting. As the tread was approached, the strains and hence the stresses became compressive and of increasing intensity, reaching a maximum at a point whose radius was equal to the mean radius of the core, beyond which they again decreased.

On the inner face over the region investigated, the strains or stresses in the radial direction were compressive, reaching the maximum nearest the bore and decreasing to a minimum at a point whose radius was equal to the mean radius of the core, after which they again increased.

(5) In the regions of the chaplets and core holes, pressing the wheel on the axle causes tensile strains in a tangential direction which are of lesser intensity than, but approach in magnitude to, those at the bore.

(6) The stresses and strains in the brackets which are produced in a radial direction by mounting are relatively insignificant.

(7) The strains caused by mounting the wheels on the axles, when mounting alone is considered, are greatest in the hub near the axles. These strains, although apparently high in the case of the greatest values recorded, are steady and not repeated as is the case with the majority of strains produced in service. Moreover, these highest strains extend through a comparatively thin layer of metal near the axle, and this strained layer is backed by other layers of less strained metal.

(8) In general the static load is transmitted from hub to rail, mainly through the outer plate, while the smaller portion of the load goes through the inner plate. This effect is more pronounced in the 740-lb Arch Plate than in the 725-lb. M. C. B. type of wheel. This division of the load seems desirable in that

the inner plate may be considered as affording reserve capacity for the purpose of absorbing the effect of side thrust on the flange when rounding curves.

(9) Pressing the wheel on the axle is much more effective in producing stress or strain within the wheel than the normal static load, and it therefore follows that the addition of the normal static load does not greatly add to or otherwise modify the more important of the existent strains caused by mounting.

(10) Abnormally heavy loads, in the absence of impact, side thrust, etc., may be sustained by wheels without increasing the normal strains, already existent, to such an extent as to seriously stress the wheel.

(11) The maximum strains reported, caused by the combined effects of mounting and static load, appear large when expressed in terms of the stress that would exist if the material were subjected to simple tension or simple compression. As previously stated, these strains are produced in the main by the mounting load, and the more important strains are those of tension, which in general are greatest near the bore of the wheel. The character of the strains, and the backing of the material most strained by less strained material, probably makes possible without injury to the material greater strains or deformations than would be allowable in the case of material not so supported and subjected to simple tensile stress. As previously stated, the problem is one of compound stress, and the method used in computing the stresses reported is thought to give the highest values for these that could be expected under such conditions. Any error in estimating, from the stress values determined in this way, when elastic failure might take place, would be upon the side of safety. In this connection it should also be remembered that the stresses reported are those produced by two forms of wheel loading only, and that the strains and stresses resulting from these two forms of loading (mounting and static) may be materially modified by additional stress-producing factors to which a wheel may be subjected in service.

APPENDIX A

A SHORT HISTORY OF THE CHILLED IRON CAR WHEEL WITH INFORMATION CONCERNING ITS MANUFACTURE AND SPECIAL PROPERTIES

If molten iron comes in contact with iron at room temperatures or is poured into a metallic mold it quickly solidifies. The sudden solidification of the metal prevents the combined carbon therein from precipitating out, thereby producing a metal in which all the carbon is in the combined form; that is, the so-called "white cast-iron" or "chilled iron" is produced. This principle was discovered in England about 1820 by a foundry workman who, quite by accident, noticed that when the excess metal remaining in the ladles after pouring the mold was emptied on the floors, the metal which came into contact with iron had quite different properties from that which solidified in contact with sand and air alone. The extreme hardness and resistance to wear of that portion of a casting chilled in this way were quite evident. In America the application of the principle of chilling to car wheel manufacture occurred about 1835. Three years later a Mr. Lobdell designed a wheel which marked the successful entrance of chilled iron into the car wheel field. In 1850 Mr. N. Washburn conceived a design of a chilled iron wheel which at that time weighed 500 lbs. The general features of this type of wheel, Figs. 42 and 43, are the merging of inner and outer plates into a single plate, the brackets and the "ogee" curve being formed on the outer face of the wheel. In 1904 the Master Car Builders Association adopted three designs of chilled wheels of the Washburn type weighing 600, 650, and 700 lbs., as recommended practice for cars of 30, 40, and 50 ton capacity. Slight variations in details of design persisted up to 1909, after which, reasonably strict adherence to the recommended M. C. B. standards was maintained. As a result of the maximum weight mentioned in the M. C. B. specifications, these three designs became commonly known as 625-lb., 675-lb., and 725-lb. wheels. In 1917 the 675-lb. Washburn type of wheel, generally spoken of herein as the M. C. B. type, for the 40-ton car was replaced by a 700-lb. pattern of different design. This design, generally called the Arch Plate type, differs from the Washburn type in that the three curves of the "ogee" curve on the outer face of the latter have been replaced

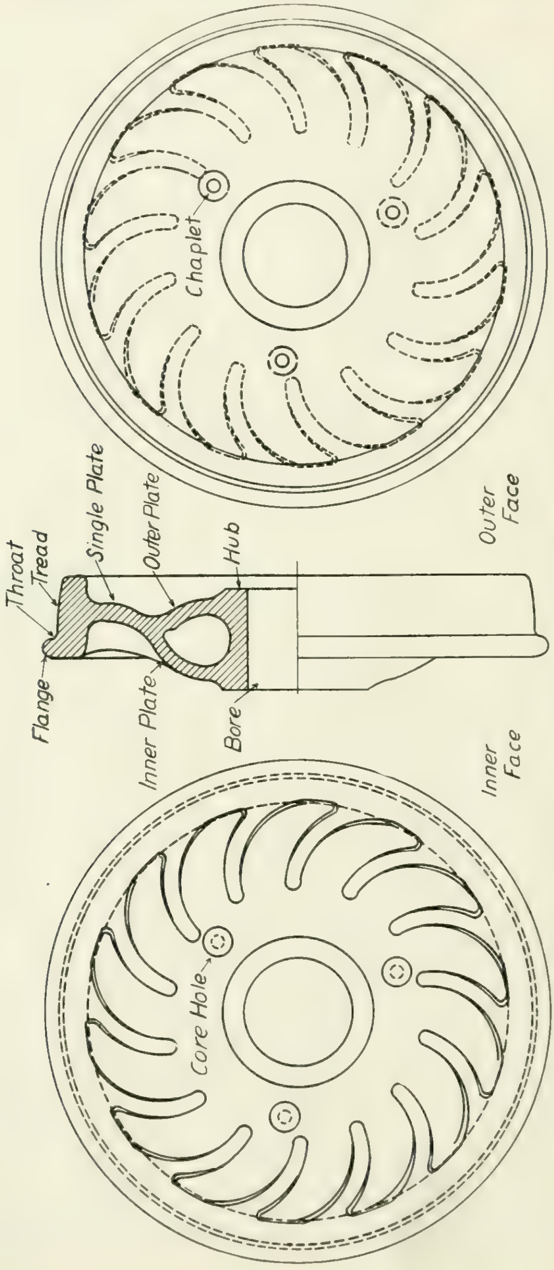
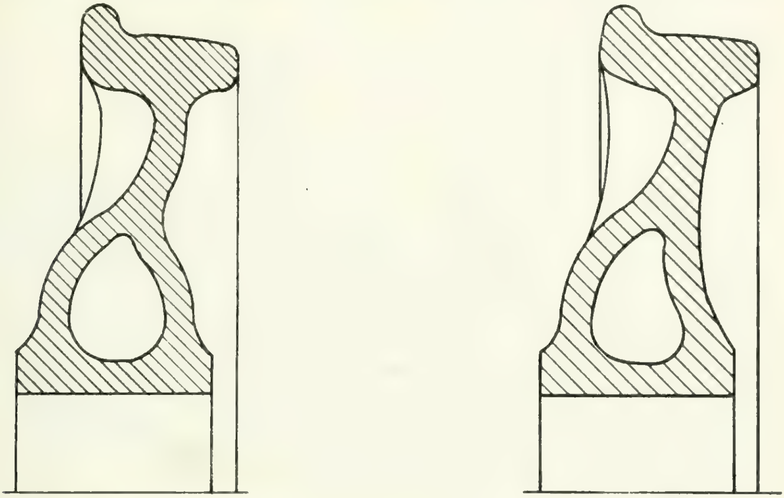


FIG. 42. M. C. B. OR WASHBURN TYPE OF CHILLED CAR WHEEL

by a single curve of large radius. An additional wheel of the Arch Plate type weighing 850-lb. was also adopted as recommended practice for 70-ton capacity cars. In 1920 the Mechanical Section of the American Railroad Association adopted as recommended practice two more car wheels of the Arch Plate design with nominal weights of 650 and 750 lbs., these to take the place of the former 625 and 725-lb. wheels. Four 33-in. cast-iron wheels all of the Arch Plate design weighing respectively 650, 700, 750, and 850 lbs. are now recommended practice for the American Railroad Association. The underlying reasons for substituting the Arch Plate for the Washburn pattern are due to both laboratory and service indications that the former, assuming equal weight of wheels in both cases, transmits the imposed loads with less internal strain, and as a corollary thereof, with a greater factor of safety. This very briefly covers the history of the chilled wheel to date.



M. C. B. or Washburn Type

Arch Plate Type

FIG. 43. SECTIONS OF CHILLED CAR WHEELS

The manufacture of chilled iron wheels is a rather complex problem and is carried on in a highly specialized manner. One of the staples of the iron markets in this country is old worn out wheels and they represent from 40 to 60 per cent of the mixture used in the production of new wheels. The balance of the material consists of pig iron, special scrap, and alloys. To give the wheel certain properties, and to enable it to meet service conditions successfully and pass the rigid inspections imposed by the recommended M. C. B. specifications,* it is necessary to control the relative proportions of scrap wheels, pig iron, and alloys quite closely. In order to do this, chemical laboratories operated by responsible chemists are maintained as adjuncts to the foundries. These laboratories furnish both the necessarily accurate control of the material that goes into the cupola and an indication of the character of the finished product. The method of molding a cast-iron car wheel is indicated by the cross-section of a car wheel mold shown in Fig. 44. Identification of the various parts of the mold is easily made by means of the figure. Attention, however, is specifically directed to the chill. This is a cast-iron ring of special chemical composition, whose inner face is machined to the desired contour of the tread and one half of the flange as shown. It rests on the nowel and in turn supports the cope of the mold. After the mold is made up, everything is ready for pouring. The molten metal is taken from the bottom of the ladle so that the impurities

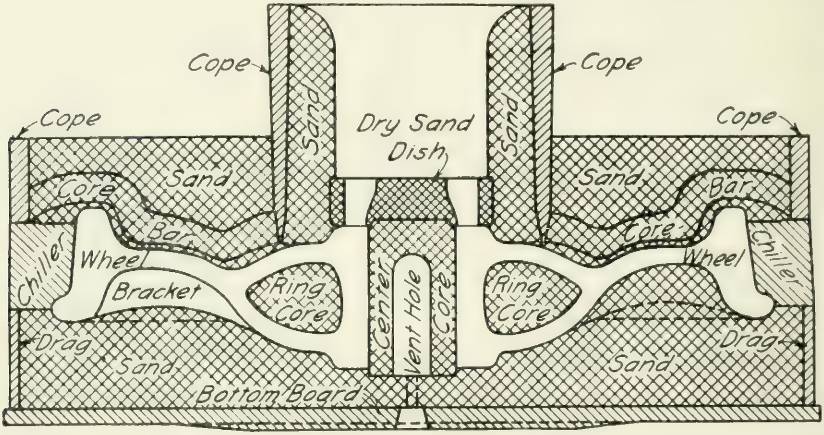


FIG. 44. CROSS SECTION OF A CAR WHEEL MOLD

remain in the ladle. About 12 seconds is taken to fill a mold for a 725-lb. wheel. When the molten metal carrying carbon in the combined state cames in contact with the cold chill, the sudden solidification of the metal does not allow sufficient time for the precipitation of the carbon from the iron—thereby producing an extremely hard metal called "chilled iron" in that part of the wheel which comes into contact with the rail. The "chilled iron" exists to a depth of from ½ to 1 in. The balance of the mold, which consists of specially tempered molding sand and two dry sand cores, allows a more gradual cooling or solidification of the metal in the other parts of the wheel. Consequently, the combined carbon in the molten metal has sufficient time to precipitate out during cooling, which in turn produces a gradation in the relative amounts of combined and graphitic carbon across the section of the wheel. This results in a wheel whose tread, or that part of the wheel which comes in contact with the rail, consists of an extremely hard metal capable of resisting both excessive deformation and wear, while the remainder of the wheel consists of relatively soft material which can be easily machined, and which is suited to resist the stresses put upon it in service.

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* Proc. M. C. B. Assn., Vol. 52, p. 490, 1918.

After the wheel has been poured it is allowed to remain in the mold for about 5 min. or until it has solidified, after which it is removed and placed in a brick lined pit. After the pits are filled they are closed, and the temperature of the wheel at first becomes uniform and then gradually reduces over a period of about 3 to 5 days. This process is called annealing, and has for its purpose the removing of such strains as may have arisen because of the unequal conditions of cooling while pouring. Proper annealing is important, as wheels cooled in air are apt to crack and are subject to failure on account of excessive internal strains. If wheels containing high residual strains were placed in service a potential danger would exist, and might result in serious wheel failures. For this reason careful attention is given to the process of pitting or annealing.

After having been cooled in the pits the wheels are removed and cleaned of molding sand by means of a sand blast, and are then subjected to rigid inspection to see if the specifications of the buyer have been fufilled. After having passed inspection the wheels are bored out at the center—the only machine work performed on the chilled car wheel during the process of manufacture. In machining, the finished diameter of the bore is made slightly smaller—about 0.002 in. per in. of bore diameter—than the axle on which it is to be mounted. The slight difference between the size of the wheel bore and the axle requires a pressure of about 10 tons per in. of axle diameter in order to force the wheel on the axle; that is, a 6-in. wheel bore would require, approximately, the pressure of about 60 tons to mount the wheel on the axle. Mounting is accomplished in specially built hydraulic wheel presses, and the wheel is then ready to be placed in service.

APPENDIX B

TABULATED RESULTS

Tables 2 to 9, inclusive, present figures relating to the strain determinations which were made in connection with the eight wheels tested. In these tables the deformations measured by the strain gage have been reduced so that they are listed as unit strains. An absence of sign indicates tensile strain; whereas a minus sign denotes compressive strain.

TABLE 2

UNIT STRAINS PRODUCED BY MOUNTING WHEEL ON AXLE
33-in. 725-lb. M. C. B. Wheel No. 671 237

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

Gage-Lines	Radials along which Gage-Lines Are Located							
	A	B	C	D	E	F	G	H
				On Outer Face of Wheel				
1R	....	—43	......	—28	......	13	......	—10
2R	....	—32	......	—109	......	—69	......	—82
3R	....	—102	......	—56	......	—127	......	—117
4R	....	—65	......	—24	......	—31	......	—12
5R	....	—105	......	—116	......	—32	......	—51
4T	....	222	......	66	......	39	......	53
5T	....	182	......	116	......	56	......	62
				On Inner Face of Wheel				
5R	....	...	....	—62	......	—64	......	—4
5T	....	108	....	177	......	173	......	218

TABLE 2A

UNIT STRAINS PRODUCED BY COMBINED EFFECTS OF MOUNTING AND STATIC LOADS

33-in. 725-lb. M. C. B. Wheel No. 671 237

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

Loads Applied at A

On Outer Face of Wheel						On Inner Face of Wheel					
Gage-Lines	Effect of Mounting Alone	Combined Effects of Mounting and Static Loads				Gage-Lines	Effect of Mounting Alone	Combined Effects of Mounting and Static Loads			
		Static Loads of						Static Loads of			
		21 000 lb.	80 500 lb.	150 000 lb.	200 000 lb.			21 000 lb.	80 500 lb.	150 000 lb.	200 000 lb.
B1R	—43	—49	—30	56	—60						
B2R	—32	—36	41	—44	—63						
B3R	—102	—99	—104	—118	—118						
B4R	—65	—68	—82	—72	—72						
B5R	—105	-131	—116	—97	—85						
B4T	222	215	213	227	217						
B5T	182	177	172	185	171	B5T	108	106	104	114	114
D1R	—28	—36	—37	......	.....						
D2R	—109	—118	—121	......	......						
D3R	—56	—65	—49	......	.....						
D4R	—24	......	......	......	...						
D5R	—116	....	......	......	......	D5R	—62	—50			
D4T	66	.....	......	......	.....						
D5T	116	....	....		...	D5T	177	161	159	182	177
F1R	13	5	16		...						
F2R	—69										
F3R	—127										
F4R	—31	—14	—30								
F5R	—32	.....	......	......	......	F5R	—64	—76	66	.	...
F4T	39	......	.....	.....	.....						
F5T	56	....	......	.....	...	F5T	173	190	195	..	...
H1R	—10	—22	—16	— 9	— 3						
H2R	—82	—99	—103	—99	—104						
H3R	—117	—125	—126	—125	—139						
H4R	—12	—11	0.0	—20	—15						
H5R	—51	—51	—43	—60	—68	H5R	—4	2	11	—7	—12
H4T	53	54	39	45	32						
H5T	62	56	53	60	65	H5T	218	210	213	221	226

TABLE 2a—Continued

UNIT STRAINS PRODUCED BY COMBINED EFFECTS OF MOUNTING AND STATIC LOADS

33-in. 725-lb. M. C. B. Wheel No. 671 237

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

Loads Applied at C

Gage-Lines	Effect of Mounting Alone	Static Loads of	
		80 500 lb.	202 500 lb.
B1R	—43	39	—35
B2R	—32	.....	.....
B3R	—102	—124	—149
B4R	—65	—76	—89
B5R	—105	—104	—108
B4T	223	233	234
B5T	182	178	177
D1R	—28	—28	—45
D2R	—109	—118	—118
D3R	—56	—66	—92
D4R	—24	—34	.....
D5R	—116	—124	—121
D4T	66	44	37
D5T	117	110	115

TABLE 3

UNIT STRAINS PRODUCED BY MOUNTING WHEEL ON AXLE
33-in. 725-lb. M. C. B. Wheel No. 671 449

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

On Outer Face of Wheel

Gage-Lines	Radials along which Gage-Lines Are Located							
	A	B	C	D	E	F	G	H
1R	—13	...	—52	—42	...	..	28	—25
2R	....	...	—67	....	...	..	—57	—59
3R	—72	...	—80	—75	...	..	—74	—116
4R	31	...	— 7	24	...	..	—49	—29
1T	20	...	41	...	...	..	5	23
2T	21	...	31	47	...	..	9	28
3T	48	...	26	37	...	..	39	44
4T	25	...	73	78	...	..	64	55
5T	....	103	....	115	...	92	..	149

On Inner Face of Wheel

1R	....	....	—16	....	....	....	....	....
1Rb	—27	—18	....	....	—29	—20	....	—6
2R	....	....	....	....	....	...	—14	....
2Rb	—53	— 6	....	....	—74	....	....	—28
3R	—23	....	....	—25	—30	0.0	....	—17
4R	—16	....	—7	4	22	—10	....	0.0
5R	....	....	—31	....	—39	....	—32	....
2T	...	...	36	..	..	..	24	..
3T	55	...	30	62	32	49	...	72
4T	110	...	57	51	15	79	...	59
5T	161	...	136	..	141	..	158	..

TABLE 3A

UNIT STRAINS PRODUCED BY COMBINED EFFECTS OF MOUNTING AND STATIC LOADS

33-in. 725-lb. M. C. B. Wheel No. 671 449
Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

Loads Applied at A

On Outer Face of Wheel						On Inner Face of Wheel					
Gage-Lines	Effect of Mounting Alone	Combined Effects of Mounting and Static Loads				Gage-Lines	Effect of Mounting Alone	Combined Effects of Mounting and Static Loads			
		Static Loads of						Static Loads of			
		21 000 lb.	80 500 lb.	150 000 lb.	200 000 lb.			21 000 lb.	80 500 lb.	150 000 lb.	200 000 lb.
A1R	—13	—19	—38	—83	—123	A1Rb	—27	—27	—37	—55	—81
A2R	. . .	. . . .	. . . .	. . . .	. . . .	A2Rb	—53	—53	—50	—61	—97
A3R	—72	—84	—140	—96	—160	A3R	—23	—28	—31	—34	—40
A4R	31	27	—15	4	—41	A4R	—16	— 9	— 7	—17	—22
A1T	20	26	42	62	95						
A2T	21	27	35	55	81		55	52	54	83	108
A3T	48	61	71	80	123	A3T	110	103	116	122	154
A4T	25	37	45	75	108	A4T	161	159	156	175	185
						A5T					

Loads Applied at G

		Static Loads of				Static Loads of	
		80 500 lb.	202 000 lb.			80 500 lb.	202 000 lb.
G1R	28	— 8	— 59		—14	2	—27
G2R	--57	—121	—197	G2R			
G3R	-74	—143	—267				
G4R	—49	— 75	—109				
G1T	5	29	82		24	37	84
G2T	9	27	74	G2T	158	164	184
G3T	39	66	115				
G4T	64	92	132				
				G5T			

TABLE 4

UNIT STRAINS PRODUCED BY MOUNTING WHEEL ON AXLE

33 in. 740-lb. Arch Plate Wheel No. 04 474

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

On Outer Face of Wheel

Gage-Lines	Radials along which Gage-Lines Are Located								Mean Experimental Strain
	A	B	C	D	E	F	G	H	
2R	48	—27	8	—14	38	— 7	52	43	18
3R	19	—34	—32	—35	—38	—18	— 2	—17	—24
4R	—80	—71	73	82	83	82	—91	—109	—84
5R	76	—20	—129	—80	—116	—83	—72	—98	—84
6R	—38	61	—57	—42	—83	—60	—70	66	—60
7R	11	—29	—72	—42	56	2	—57	—35	—38
8R	2	8	—17	—12	—23	— 9	—24	20	—14
9R	3	26	—17	8	— 9	—24	—18	— 6	— 7
1T	77	74	95	105	172	137	68	78	101
2T	68	113	109	67	142	127	143	97	108
3T	68	155	85	23	70	97	19	85	75
4T	52	115	36	17	65	63	35	45	54
5T	25	71	28	26	45	37	61		42
6T	38	35	16	23	23	7	36	34	26
7T	16	13	— 3	8	13	26	27	18	15
8T	23	11	—14	3	18	2	— 6	48	11
9T	11	26	— 4	25	20	19	8	4	14

On Inner Face of Wheel

Gage-Lines	A	B	C	D	E	F	G	H	Mean Experimental Strain
2R	—39	—23	—66	—52	69	—38	—47	—13	—43
3R	—30		0 0	—29	42	15	17	17	— 7
4R	— 8		— 4	—16	4	— 8	3	4	— 4
5R	21		— 1	— 6	1	—29	—15	7	— 9
6Ra, b				. . .		—52	—58	—59	—56
6R			. .	—18	2		—29		—15
7Ra	—34	44	23	. . .			28	—42	—34
7R	—38	12	—19	—34	—29	—64	—47	—57	—34
7Rb			—33	—26	—38	— 3			—25
9Ra	—26	—11	39	. .			8	—17	—17
9Rb	....		—17	—27	—32	—19			24
1T	174	263	200	171	200	150	197	197	194
2T	95	282	187	128	164	91	149	139	154
3T	82		131	77	44	98	48	80	80
4T	81		105	74	30	72		96	76
5T	51		98	70	41	52	63	74	64
6T	....		84	....	....	42		....	63

TABLE 4A

UNIT STRAINS PRODUCED BY COMBINED EFFECTS OF MOUNTING AND STATIC LOADS

33-in. 740-lb. Arch Plate Wheel No. 04 474

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

Loads Applied at A

On Outer Face of Wheel

Gage-Lines	Effect of Mounting Alone	Combined Effects of Mounting and Static Loads — Static Loads of 21 100 lb.	81 000 lb.	150 000 lb.	202 000 lb.
A2R	48	16	14	59	56
A3R	—19	—33	—90	—65	—89
A4R	—80	—105	—157	—176	-249
A5R	—76	—96	—185	—167	—220
A6R	—38	—61	—124	—125	—174
A7R	—11	—29	—76	—80	-113
A8R	2	—35	—14	—70	—109
A9R	3	— 9	—31	—60	—94
A1T	77	54	113	110	127
A2T	68	76	117	102	109
A3T	68	34	104	88	76
A4T	52	36	75	64	91
A5T	25	11	42	33	56
A6T	38	27	48	56	74
A7T	16	7	23	32	37
A8T	23	7	22	52	93
A9T	11	9	25	39	63

On Inner Face of Wheel

Gage-Lines	Effect of Mounting Alone	Combined Effects of Mounting and Static Loads — Static Loads of 21 100 lb.	81 000 lb.	150 000 lb.	202 000 lb.
A2R	—39	—42	—35	—45	—72
A3R	—30	—25	— 6	—35	—62
A4R	— 8	—15	— 2	0.0	—30
A5R	—21	—27	—27	—32	—27
A7Ra	—34	—33	—15	—44	—65
A7R	—38	—37	—18	—57	—65
A9Ra	—26	—42	—35	—80	—107
A1T	174	175	179	189	203
A2T	95	89	130	127	135
A3T	82	110	105	120	131
A4T	81	67	90	123	142
A5T	51	57	58	67	125

Loads Applied at C

On Outer Face of Wheel

Gage-Lines	Effect of Mounting Alone	Static Loads of 21 400 lb.	81 500 lb.	151 000 lb.	200 000 lb.
C2R	8	27	3	4	3
C3R	—32	—78	—111	—109	—116
C4R	—73	—87	—155	—162	—194
C5R	—129	—120	—179	—214	—251
C6R	—57	—61	—113	—131	—155
C7R	—72	—71	—135	—115	—150
C8R	—17	—37	—99	—56	—89
C9R	—17	—18	—60	—70	—94
C1T	95	81	128	123	149
C2T	109	87	130	145	147
C3T	85	36	100	127	145
C4T	36	29	57	...	40
C5T	28	27	41	47	53
C6T	16	29	39	51	68
C7T	— 3	1	15	20	37
C8T	—14	0.0	24	27	54
C9T	— 4	...	20	38	79

On Inner Face of Wheel

Gage-Lines	Effect of Mounting Alone	Static Loads of 21 400 lb.	81 500 lb.	151 000 lb.	200 000 lb.
C2R	—66	—55	—49	—70	—83
C3R	0.0	12	20	— 5	—13
C4R	— 4	—10	— 2	2	10
C5R	— 1	—12	0.0	5	13
C7Ra	—23	—19	— 1	—42	—42
C7R	—19	— 7	9	—19	—24
C7Rb	—33	— 6	3	—19	— 9
C9Ra	—39	—18	—22	—98	—112
C9Rb	—17	—18	— 3	—18	—16
C1T	200	197	180	190	205
C2T	187	169	171	182	200
C3T	131	132	118	141	159
C4T	105	116	82	133	133
C5T	98	75	84	106	136
C6T	84	73	76	122	142

TABLE 5

UNIT STRAINS PRODUCED BY MOUNTING WHEEL ON AXLE

33 in. 740-lb. Arch Plate Wheel No. 04-476

Strains Given in Table are Expressed in Hundred thousandths of Inches Per Inch

On Outer Face of Wheel

Gage-Lines	Radials along which Gage-Lines Are Located								Mean Experimental Strain
	A	B	C	D	E	F	G	H	
2R	36	4	20	—81	—22	—29	— 3	46	15
3R	—15	—61	—31	—23	—61	—49	—61	—36	42
4R	—64	—88	—68	—77	112	—86	—128	—58	85
5R	71	90	—96	—114	—89	—82	105	—74	90
6R	—38	65	—47	53	—53	—67	—71	53	56
7R	23	24	—23	45	45	.	—35	—18	30
8R	—11	31	— 9	12	3	—14	—12		13
9R	34	3	0.0	8	— 1	—23	—18	41	6
1T	123	114	68	129	114	102	83	68	100
2T	128	80	79	70	98	44	99	94	86
3T	59	137	67	51	53	62	37	75	68
4T	33	90	40	16	30	37	123	42	51
5T	36	46	19	22	13	22	49	26	29
6T	26	32	47	23	13	8	.	18	24
7T	29	23	8	—39	14	— 3	7	25	4
8T	15	5	11	43	— 1	11	11	20	14
9T	— 3	— 4	2	—14	15	— 7	27	2	2

On Inner Face of Wheel

Gage-Lines	A	B	C	D	E	F	G	H	Mean Experimental Strain
2R	—40	—41	—50	—99	—88	—80	—42	—32	—59
3R	7	....	—14	—23	—23	—18	—35	—10	—17
4R	— 1	.	—30	— 8	—15	11	— 7	—12	— 9
5R	28	...	—22	—23	—24	—41	—21	—25	26
6Ra				.	...		...	—23	
6R		..	—29	..	37	—42	6	.	—26
7Ra	—42	—40	—35	—35	—41	—61	53	32	—29
7R		.	..	—42	—39	—31	—28	..	—35
7Rb				...	..	—41	....		
8R				..	..	—15	...		..
9Ra	27	— 8	—31	—19	—43	—48	—21	0.0	—25
9Rb		.				—37		.	..
1T	174	167	160	195	166	123	176	180	168
2T	112	189	133	129	128	127	172	117	138
3T	84		97	78	68	98	....	70	82
4T	60		53	61	21	87		83	61
5T	55		54	74	47	68	.	81	63
6T	35		36	85	65	25	71	35	50

Table 5a

Unit Strains Produced by Combined Effects of Mounting and Static Loads

33-in. 740-lb. Arch Plate Wheel No. 04 476

Strains Given in Table are Expressed in Hundred thousandths of Inches Per Inch

Loads Applied at A

On Outer Face of Wheel						On Inner Face of Wheel					
		Combined Effects of Mounting and Static Loads						Combined Effects of Mounting and Static Loads			
Gage-Lines	Effect of Mounting Alone	Static Loads of				Gage-Lines	Effect of Mounting Alone	Static Loads of			
		21 100 lb.	81 000 lb.	150 000 lb.	202 000 lb.			21 100 lb.	81 000 lb.	150 000 lb.	202 000 lb.
A2R	36	23	— 4	21	26	A2R	—40	—41	—32	—66	—80
A3R	—15	—33	—92	—68	—92	A3R	7	5	14	— 5	—13
A4R	—64	—95	—138	—147	—200	A4R	— 1	12	14	2	— 1
A5R	—71	—94	—143	—169	—213	A5R	—28	—32	—15	31	—104
A6R	—38	—62	—102	—111	—151						
A7R	—23	—38	—66	—88	118	A7Ra	42	29	19	65	76
A8R	—11	—20	—55	—87	—122						
A9R	34	13	—35	—41	83	A9Ra	—27	—22	—38	—90	111
A1T	123	116	156	145	163	A1T	174	178	202	196	198
A2T	128	150	181	155	173	A2T	112	144	123	134	151
A3T	59	54	72	83	97	A3T	84	74	91	122	140
A4T	33	30	41	43	71	A4T	60	92	86	79	107
A5T	36	35	29	28	50	A5T	55	43	67	93	101
A6T	26	10	18	41	59	A6T	35	23	53	62	81
A7T	29	30	51	45	109						
A8T	15	3	32	32	57						
A9T	—3	—25	2	26	65						

Loads Applied at G

		Static Loads of						Static Loads of			
		21 400 lb.	81 500 lb.	151 000 lb.	200 000 lb.			21 400 lb.	81 500 lb.	151 000 lb.	200 000 lb.
G2R	— 3	18	4	—11	— 6	G2R	—42	—82	—66	—90	—90
G3R	—61	—52	—103	—114	—137	G3R	35			—69	—72
G4R	—128	111	—177	—211	—262	G4R	— 7	—51	— 4	12	23
G5R	—105	111	—166	—186	—238	G5R	—21	—31	—14	—29	—16
G6R	—71	—87	—121	—106	—154	G6R	6	—25	—11	2	6
G7R	—35	—39	—75	—69	—107	G7Ra	53	51	82	50	69
G8R	—12	0.0	—38	—62	—104	G7R	—28	—25	—17	—32	—19
G9R	—18	35	—17	—58	—109	G9Ra	—21	—29	— 3	—48	—42
G1T	83	71	106	122	140	G1T	176	174	193	187	159
G2T	99	105	148	94	157	G2T	172	166	200	186	219
G3T	37	29	60	38	65	G6T	71	69	95	84	119
G4T	123	113	118	135	172						
G5T	49	35	19	49	48						
G7T	7	14	30	24	30						
G8T	11	—1	11	21	16						
G9T	27	6	32	63	94						

TABLE 6

UNIT STRAINS PRODUCED BY MOUNTING WHEEL ON AXLE

33-in. 625-lb. M. C. B. Wheel No. 809 711

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

On Outer Face of Wheel						On Inner Face of Wheel					
Gage-Lines	Radials along which Gage-Lines Are Located				Mean Experimental Strain	Gage-Lines	Radials along which Gage-Lines Are Located				Mean Experimental Strain
	A	B	C	D			A	B	C	D	
At	37	115	95	64	78	At	115	190	142	126	143
Bt	67	109	. .	83	86	Bt	100	163	137	102	126
Ct	50	68	58	83	65	Ct	92	136	120	103	113
Dt	40	108	65	86	75	Dt	69	118	100	81	92
Et	37	81	51	60	57	Et	64	107	85	71	82
Ft	18	. .	67	65	50	. . . .	. .	. .	. .	. .	. .
Gt	28	58	38	70	48	. . . .	. .	. .	. .	. .	. .
2T	20	57	50	. .	42	2T	50	75	53	58	59
3T	20	43	28	35	32	3T	46	50	26	39	40
4T	19	9	18	20	16	4T	30	20	9	. .	20
5T	5	25	18	11	15	5T	18	31	24	. .	24
6T	4	22	— 6	— 3	4	6T	27	2	28	– 6	13
7T	9	9	11	1	8	6Tab	. .	16	. .	8	12
8T	15	15	3	— 8	6	7T	18	28	23	8	19
9T	—14	1	— 6	11	— 2	. . . .	. .	. .	. .	. .	. .
10T	— 6	— 1	4	15	3	. . . .	. .	. .	. .	. .	. .
1R	—20	—53	—17	—30	—30	1R	—37	31	—56	—32	—24
2R	—23	—35	—31	—20	—27	2R	—31	—33	—29	—38	—33
3R	—26	—18	—27	— 5	—19	3R	— 9	. .	—27	—25	—20
4R	—15	—23	—39	—20	—24	4R	—29	—24	—13	— 7	—18
5R	—17	—10	—33	—25	—21	5R	— 5	— 2	0.0	—20	— 7
6R	—40	—12	— 6	—19	—19	6R	—10	— 8	—28	1	—11
7R	—19	—19	—24	—21	—21	6Rab	. .	—25	. .	—46	—36
8R	— 6	0.0	—20	— 2	— 7	7R	—15	—13	— 6	—16	—12
9R	— 5	– 1	—17	— 5	— 7	8Rab	25	7	8	—18	6
						9Rab	9	. .	3	—13	0.0

	Gage-Lines Adjacent to Core Holes			
	Between Radials A and B	Between Radials B and C	Between Radials D and A	
1T	85	72	68	75
2T	49	111	88	83
3T	118	66	. .	92
4T	118	46		82

TABLE 7

UNIT STRAINS PRODUCED BY MOUNTING WHEEL ON AXLE

33-in. 625-lb. M. C. B. Wheel No. 822 269

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

On Outer Face of Wheel						On Inner Face of Wheel					
Gage-Lines	Radials along which Gage-Lines Are Located				Mean Experi-mental Strain	Gage-Lines	Radials along which Gage-Lines Are Located				Mean Experi-mental Strain
	A	B	C	D			A	B	C	D	
At	161	213	343	150	217	At	192	156	161	245	188
Bt	127	171	224	130	163	Bt	148	131	165	228	168
Ct	107	161	249	124	160	Ct	136	105	140	196	144
Dt	115	137	171	91	128	Dt	127	97	129	179	133
Et	97	115	174	106	123	Et	106	90	131	174	125
Ft	39	121	140	84	96	....	..	..	..	..	...
Gt	95	99	140	94	107	....	..	..	..	..	....
2T	86	93	128	76	96	2T	104	77	118	147	112
3T	60	59	105	53	69	3T	84	62	79	148	93
4T	..	54	71	45	57	4T	45	51	39	..	45
5T	40	37	44	33	38	5T	38	45	38	..	40
6T	37	41	16	16	28	6T	29	16	28	51	31
7T	26	16	31	12	21	6Tb	..	—21	..	..	....
8T	9	6	11	9	9	7T	29	12	43	42	32
9T	20	2	22	14	14	....	..	..	..	..	....
10T	11	3	15	— 4	6	....	..	..	..	..	....
1R	—50	—88	..	—47	—62	1R	—29	—53	—58	—66	—52
2R	—42	—50	—77	—56	—56	2R	..	—56	—34	—54	—48
3R	— 9	—33	—47	—31	—30	3R	—22	—20	—29	—38	—27
4R	—38	—44	—50	—36	—42	4R	—24	—20	—10	— 9	—16
5R	—40	—54	—57	—55	—51	5R	—16	—24	— 4	—21	—16
6R	—59	—64	—52	—55	—58	6R	— 6	—24	—16	..	—15
7R	—52	—37	—37	—49	—44	6Rab	..	—24	..	—33	—28
8R	—18	—29	—16	—18	—20	7R	—11	—16	—20	—13	—15
9R	—41	—16	— 4	9	—13	8Rab	10	0.0	34	15	15
						9Rab	4	24	10	-10	7

	Gage-Lines Adjacent to Core Holes			
	Between Radials A and B	Between Radials B and C	Between Radials D and A	
1T	87	100	126	104
2T	130	118	....	124
3T	....	101	....	101
4T	65	57	....	61

TABLE 8

UNIT STRAINS PRODUCED BY MOUNTING WHEEL ON AXLE
33 in. 725-lb. M. C. B. Wheel No. 49 317

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

	On Outer Face of Wheel						On Inner Face of Wheel				
Gage-Lines	Radials along which Gage-Lines Are Located				Mean Experimental Strain	Gage-Lines	Radials along which Gage-Lines Are Located				Mean Experimental Strain
	A	B	C	D			A	B	C	D	
At	81	79	152	122	108	At	134	129	95	109	117
Bt	68	79	134	121	100	Bt	114	113	86	100	103
Ct	59	69	125	69	80	Ct	107	80	79	90	89
Dt	47	67	94	99	77	Dt	99	87	77	73	84
Et	48	61	88	94	73	Et	92	84	81	73	82
Ft	44	64	101	96	76	Ft	83	76	67	72	74
Gt	29	60	76	93	64	Gt	83	68		66	72
3T	38	40	56	70	51	3T	60	51	49	65	56
4T	28	34	36	51	37	4T	51	37	33		40
5T	34	28	24	36	30	5T	17	22	21		20
6T	22	24	26	28	25	6T	23	44	21	.	29
7T	5	17	21	21	16	7T	14	19	18	.	17
8T	10	7	8	13	10				.	.	..
9T	10	8	14	11	11			.	.		..
10T	6	8	3	7	6				.	..	..
1R	—51	—39	—49	—57	—49	1R	—57	—82	—54	—55	—62
2R	—22	—23	—47	—47	—35	2R	—30	—31	—35	—35	—33
3R	—23	—23	—26	—36	—27	3R	—17	—22	—22	—25	—22
4R	—17	—16	—19	—33	—21	4R	— 6	— 8	— 7	—10	— 8
5R	—14	—23	—22	—35	—24	5R	— 4	— 5	7	—10	— 3
6R	—35	—34	—40	—51	—40	6R	— 6	— 1	— 7	— 2	— 4
7R	—28	—29	—39	—44	—35	7R	—19	—17	—20	—19	—19
8R	—13	—17	—17	—28	—19	8R	2	..	—27		—12
9R	1	— 8	—11	—11	— 7	8Ra	— 6	—10	—13	4	— 6
						9Ra	— 8	— 9	—10	2	— 6

	Gage-Lines Adjacent to Chaplets					Gage-Lines Adjacent to Core Holes			
	Between Radials A and B	Between Radials B and C	Between Radials D and A			Between Radials A and B	Between Radials B and C	Between Radials D and A	
1T	34	63	64	54	1T	51	45	54	50
2T	45	70	75	63	2T	63	53	80	65
3T	48	82	69	66	3T	63	40	47	50
4T	47	60	51	53	4T	24	15	13	17
5T	32	36	50	39					

TABLE 9

UNIT STRAINS PRODUCED BY MOUNTING WHEEL ON AXLE

33 in. 725-lb. M. C. B. Wheel No. 49 305

Strains Given in Table are Expressed in Hundred-thousandths of Inches Per Inch

	On Outer Face of Wheel						On Inner Face of Wheel				
Gage-Lines	Radials along which Gage-Lines Are Located				Mean Experimental Strain	Gage-Lines	Radials along which Gage-Lines Are Located				Mean Experimental Strain
	A	B	C	D			A	B	C	D	
At	297	104	87	139	157	At	97	167	149	156	142
Bt	268	82	67	116	133	Bt	91	132	125	126	118
Ct	240	79	61	113	123	Ct	83	114	116	110	106
Dt	208	47	49	110	104	Dt		129	107	95	110
Et	148	62	56	109	94	Et	74	119	96	80	92
Ft	134	68	44	110	89	Ft	76	100	89	80	86
Gt	128	68	44	109	87	Gt	65	94	83	65	77
3T	74	40	51	97	66	3T	48	61	65	65	60
4T	55	36	36	55	46	4T	39	42	22		34
5T	36	26	33	.	32	5T	46	48	11	0 0	35
6T	26	58	30	65	45	6T	6	14	21		14
7T	13	15	20	24	18	7T	8	30	20		19
8T	12	6	15	18	13	. .		..	.		. .
9T	14	7	15	12	12	.					
10T	3	7	14	20	10	. .	. .			. .	.
1R	—209	—114	—30	—56	—102	1R	—72	—110	—82	—197	—115
2R	—155	— 89	— 1	—52	— 74	2R	—59	— 60	—69	—115	— 76
3R	— 94	— 28	2	.	— 40	3R	—35	— 32	—44	—104	— 54
4R	— 65	— 22	— 3	—30	— 30	4R	—23	— 25	—27	— 82	— 39
5R	—108	34	—20	— 32	— 48	5R	10	— 18	—30	— 51	— 27
6R	— 50	— 32	—40	—79	— 50	6R	—31	— 22	—16	— 40	— 27
7R	— 47	— 34	—41	—54	— 44	7R	—20	— 24	—36	— 18	— 24
8R	29	— 26	—22	—19	24	8R	—21	.	—20		— 20
9R	— 17	— 1	—11	9	— 10	8Ra	—12	— 8	— 8	— 8	— 9
						9Ra	7	14	—22	7	— 9

Gage-Lines Adjacent to Core Holes

	Between Radials A and B	Between Radials B and C	Between Radials D and A	
1T	65	82	69	72
2T	82	82	.	82
3T	70	73	60	68
4T	56	48	39	48

APPENDIX C

AGREEMENT BETWEEN THE ASSOCIATION OF MANUFACTURERS OF CHILLED CAR WHEELS AND THE UNIVERSITY OF ILLINOIS, PROVIDING FOR A COÖPERATIVE INVESTIGATION RELATING TO CHILLED CAR WHEELS

The following articles of agreement, prepared by the Engineering Experiment Station of the University of Illinois, were submitted to, and, on January 26, 1916, approved by Geo. W. Lyndon, President of, and acting for, the Association of Manufacturers of Chilled Car Wheels. The articles of agreement were likewise submitted to, and, on February 8, 1916, approved by the Board of Trustees of the University of Illinois.

(1) That the Engineering Experiment Station will undertake an investigation concerning the stresses and behavior of chilled iron car wheels, under prescribed conditions, the details of the work to be determined by the Station in conference with the Association's Consulting Engineer, Mr. F. K. Vial.

(2) That all results secured from such an investigation shall be the property of the Engineering Experiment Station. They may be published as Station bulletins or otherwise, as from time to time may be determined by the Director. Prior to any publication, however, the intention of the Director shall be formally communicated to the Association of Manufacturers of Chilled Car Wheels in order that said Association may at all times be informed as to the purpose of the Station. The results also shall at all times be open to the inspection and use of the representative of the Association of Manufacturers of Chilled Car Wheels, who may be assumed to act in an advisory capacity to the Station.

(3) That the funds of the Association of Manufacturers of Chilled Car Wheels are to be drawn upon for all costs of labor and supplies arising from the proposed investigation. If equipment is required in excess of that already available to the Station, it will be provided at the expense of the Association. The Station will supply the time of members of its staff for the direction of the work, and the use of such facilities as it may already possess.

(4) That upon the approval of these proposals the Association of Manufacturers of Chilled Car Wheels will at once transmit to the Business Office of the University the sum of $500 as an initial deposit, and each month thereafter will transmit such sum or sums as may be called for by estimates of the costs of the investigation for the month next succeeding.

Funds thus supplied are to constitute a credit upon which the Director of the Engineering Experiment Station may from time to time draw to cover the costs of the proposed work. It will be the purpose to maintain, during the lifetime of this agreement, a balance in this account entirely sufficient to meet all charges that will need to be made against it.

(5) That this arrangement will continue in effect until canceled through the definite action of one or both of the parties in interest.

LIST OF
PUBLICATIONS OF THE ENGINEERING EXPERIMENT STATION

Bulletin No. 1. Tests of Reinforced Concrete Beams, by Arthur N. Talbot. 1904. *None available.*

Circular No. 1. High-Speed Tool Steels, by L. P. Breckenridge. 1905. *None available.*

Bulletin No. 2. Tests of High-Speed Tool Steels on Cast Iron, by L. P. Breckenridge and Henry B. Dirks. 1905. *None available.*

Circular No. 2. Drainage of Earth Roads, by Ira O. Baker. 1906. *None available.*

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Bulletin No. 9. An Extension to the Dewey Decimal System of Classification Applied to the Engineering Industries, by L. P. Breckenridge and G. A. Goodenough. 1906. Revised Edition, 1912. *Fifty cents.*

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ENGINEERING EXPERIMENT STATION

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THE
REHEATING OF COMPRESSED AIR

BY

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THE REHEATING OF COMPRESSED AIR

I. Introduction

1. *Preliminary Statement.*—For many years compressed air has been employed as a medium for the transmission of power. The simplicity and relatively high efficiency of air compressors, the economy with which compressed air may be transmitted for comparatively long distances through pipe lines, and the variety of air motors and tools now available have led to the extensive use of compressed air in mining, quarrying, and tunneling, in various shop processes, and in the development of power.

Many devices operated by compressed air are run intermittently; and with such apparatus economy of operation is generally sacrificed for simplicity and minimum initial cost of equipment. Few air-driven tools use the air expansively, wherefore their thermal efficiency is small. When motors are operated continuously for considerable periods of time, however, economy of operation becomes a consideration, and it is essential that the air be expanded in the motor. Since the variations of volume and pressure of air during compression and expansion may be represented by the equation pv^n = a constant, air is heated during compression and cooled during expansion. In the compressed air plant the heat of compression is quickly dissipated by radiation from the air receivers and pipe lines, so that when the air reaches the motor its temperature is little if any higher than that of the atmosphere. Consequently the temperature of the air after expansion is so low that it causes serious operating difficulties, resulting both from the freezing of the moisture in the air and the consequent accumulation of ice in the exhaust pipe, and from the ineffectiveness of cylinder lubrication and resultant loss of mechanical efficiency. These difficulties may be overcome by heating the air before its delivery to the motor to an initial temperature such that the temperature of the exhaust air will be above the freezing point of water. This heating process is commonly designated as "reheating," and the term will therefore be employed in the discussion which follows.

It should be noted that when compressed air is used as a medium for the transmission of power the reheating process affords a means of increasing the total output of power per pound of air through the expenditure of a small additional amount of heat energy, and that no other system of power transmission offers similar advantages.

2. *Purposes of Investigation and Significant Results.*—This investigation of the reheating of compressed air was undertaken to determine the ideal thermodynamic efficiencies resulting from the heat expended in the reheating process, the efficiency of external and internal combustion reheaters, the performance of an engine using air expansively under a wide variety of operating conditions, and the performance of the same engine operating with steam alone, and with mixtures of air and steam when steam is injected into the air pipe as a means of reheating the air.

The results of the investigation of reheaters show that in small external combustion reheaters maximum efficiencies of from 16.7 to 61.5 per cent may be secured, depending upon the type of reheater employed; and that in the internal combustion reheater of the type tested the efficiency varied from 69.4 per cent when 326 pounds of air per hour was heated, to 83.0 per cent when 1240 pounds of air per hour was heated.

The tests of the engine using air as the working medium show that under the most favorable conditions maintained the expenditure of coke per indicated horse power per hour gained through reheating was 0.98 pounds, and, per brake horse power per hour gained, 0.90 pounds. Under the least favorable conditions of operation with cold air the mechanical efficiency of the engine fell to 59.8 per cent, while under the best conditions of operation with reheated air the mechanical efficiency was 93.7 per cent.

In the tests of the engine using a mixture of air and steam it was found that the work done per pound of mixture was considerably in excess of that attainable by the separate use of the same weight of each ingredient in the mixture. The interest in air-steam mixtures developed during this investigation has led to further studies of the subject, the results of which will be presented in a later bulletin.

3. *Acknowledgments.*—The experimental work of this investigation was carried on in the Power Laboratory of the Department of Mechanical Engineering by JOHN N. VEDDER,* Research Assistant in Mechanical Engineering. During a part of the time, he was

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* Now Associate Professor of Thermodynamics at Union College, Schenectady, New York.

assisted by WILLIAM E. MOSHER,† Research Graduate Assistant in Mechanical Engineering. Because of uncertainty regarding the accuracy of the method originally employed in calibrating the Venturi tube used as an air meter during the tests, the tube was recently recalibrated and all air weights recalculated by CRANDALL Z. ROSECRANS, Research Assistant in Mechanical Engineering. This recalibration was performed with the air weighing plant described on page 81 of Engineering Experiment Station Bulletin No. 120, entitled "Investigation of Warm Air Furnaces and Heating Systems."

The authors are indebted to the Ingersoll-Rand Company, the Sullivan Machinery Company, and the Kreutzburg Manufacturing Company for the loan of reheaters used during the investigation.

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† Mr. Mosher died in Washington, D. C., on October 12, 1918. At that time he was Assistant Superintending Engineer of the U. S. Army Transport Service, in charge of marine refrigeration.

II. Thermodynamics of Reheated Air: Ideal Cycles

4. *Case I. No Expansion.*—Compressed air may be used in motors in which there is no expansion of the air, in those in which there is partial expansion, and in those in which there is complete expansion. The ideal efficiency resulting from the use of reheated air—that is, the ratio of the extra theoretical work secured through the use of reheated air, expressed in British thermal units (B.t.u.), to the heat added to reheat the air, in B.t.u.—is here developed for these three cases.

In Fig. 1 let $p_e = p_a = p_b =$ the initial pressure in pounds per square foot of air admitted to the cylinder of a motor whose clearance

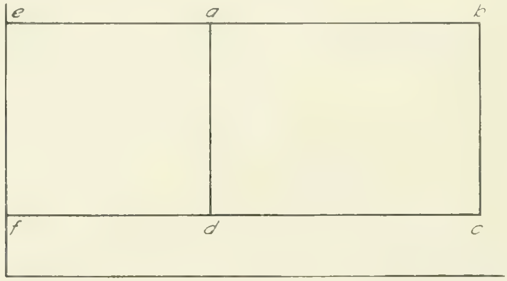

Fig. 1. Work Diagram for Air. No Expansion

volume is zero; and $p_f = p_d = p_c =$ the pressure of the atmosphere. Assume that $v_a =$ the volume in cubic feet of one pound of air whose absolute temperature is T_a. The theoretical work in foot-pounds per pound of air at T_a is represented by the area $e\ a\ d\ f$. If the air is heated at constant pressure to a temperature T_b its volume will become v_b and the theoretical work per pound of air is represented by the area $e\ b\ c\ f$. The difference between these areas, $a\ b\ c\ d$, represents the additional work per pound of air resulting from the heat added as "reheat."

$$\text{The area } a\,b\,c\,d = (p_a - p_d)(v_b - v_a) \text{ ft. lbs.}$$
$$= A\,(p_a - p_d)(v_b - v_a) \text{ B.t.u.}$$

when $\quad A = \dfrac{1}{778}$ the reciprocal of the mechanical equivalent of heat.

The heat supplied $= c_p\,(T_b - T_a)$ B.t.u.

$$= c_p\,\frac{T_a}{v_a}\,(v_b - v_a)\ \text{B.t.u.,}$$

since
$$\frac{T_a}{v_a} = \frac{T_b}{v_b}$$

and $\quad c_p$ = mean specific heat of air at constant pressure.

The efficiency ratio then becomes

$$E = \frac{A\,(p_b - p_a)\,(v_b - v_a)}{c_p\,\dfrac{T_a}{v_a}\,(v_b - v_a)} = \frac{A\,v_a\,(p_b - p_a)}{c_p\,T_a}$$

$$\frac{p_a\,v_a}{T_a} = B, \text{ the air constant.}$$

If $\quad c_v$ = the mean specific heat of air at constant volume

$$B = \frac{c_p - c_v}{A}$$

and
$$\frac{v_a}{T_a} = \frac{B}{p_a} = \frac{c_p - c_v}{A\,p_a}$$

Hence
$$E = \frac{k-1}{k}\left(1 - \frac{p_a}{p_b}\right) = 0.2884\left(1 - \frac{p_a}{p_b}\right)$$

since
$$k = \frac{c_p}{c_v} = 1.405$$

5. *Case II. Complete Expansion.*—In Fig. 2 let $p_b = p_c = p_f$ = the initial pressure; $p_a = p_d = p_e$ = the atmospheric pressure in pounds per square foot; and v_a = the volume in cubic feet of one pound of air at a temperature T_a and pressure p_a. If in a cylinder without clearance the air be expanded to the atmospheric pressure p_d, the work in foot pounds for the cycle is represented by the area

e a d f. If the air is reheated at constant pressure from T_a to T_b, its volume is increased from v_a to v_b; and the work of the cycle for complete expansion is represented by the area *e b c f*. The difference between the areas, *a b c d*, represents the additional work per pound of air secured through the heat added. The efficiency of the reheating process is the ratio of the heat converted into work to the heat added. With complete expansion of the air, it is obvious that the efficiency is the maximum attainable.

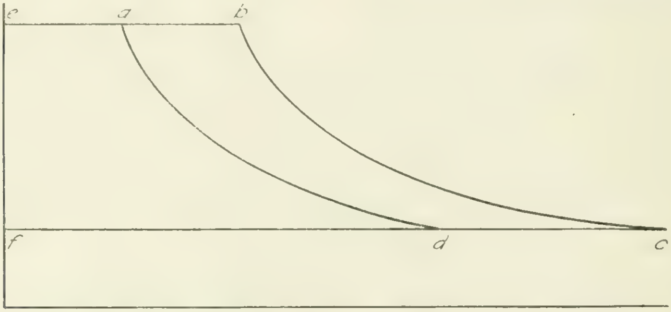

Fig. 2. Work Diagram for Air. Complete Expansion

$$\text{Since } p_a v_a^n = p_d v_d^n$$
$$\text{and } p_b v_b^n = p_c v_c^n$$

$$\text{the work area } e\ a\ d\ f = \frac{n}{n-1} p_a v_a \left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right] \text{ft. lbs.,}$$

$$\text{and the work area } e\ b\ c\ f = \frac{n}{n-1} p_b v_b \left[1 - \left(\frac{p_c}{p_b}\right)^{\frac{n-1}{n}}\right] \text{ft. lbs.,}$$

$$= \frac{n}{n-1} p_a v_b \left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right] \text{ft. lbs.}$$

$$\text{The work gained } a\ b\ c\ d = \frac{n}{n-1} p_a (v_b - v_a) \left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right] \text{ft. lbs.,}$$

$$= \frac{n}{n-1} B \left(T_b - T_a\right) \left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right] \text{ft. lbs.}$$

$$\text{The ``reheat'' supplied } = \frac{1}{A} c_p \left(T_b - T_a\right) \text{ft. lbs.}$$

The efficiency then becomes

$$E = \frac{n}{n-1}\left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right]\frac{A\ B}{c_p}$$

$$= \left(\frac{n}{n-1}\right)\left(\frac{k-1}{k}\right)\left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right]$$

$$= \left(\frac{n}{n-1}\right)\left(\frac{k-1}{k}\right)\left[1 - \left(\frac{v_a}{v_d}\right)^{n-1}\right]$$

$$= \left(\frac{n}{n-1}\right)\left(\frac{k-1}{k}\right)\left[1 - \frac{T_d}{T_a}\right]$$

If the air is expanded adiabatically n becomes equal to k and the efficiency is reduced to

$$E = 1 - \left(\frac{p_d}{p_a}\right)^{\frac{k-1}{k}} = 1 - \left(\frac{v_a}{v_d}\right)^{k-1} = 1 - \frac{T_d}{T_a}$$

6. *Case III. Partial Expansion.*—The least favorable case for reheated air used expansively is shown by the diagram, Fig. 3. As in Case II, the area $e\ a\ d\ f$ represents the work in foot pounds per pound of air whose initial volume is v_a at a pressure p_a and temperature T_a. Assume that the temperature is increased to T_b at constant pressure, and that the volume v_c at the end of expansion is the same as the volume v_d attained after expansion of the unreheated air to atmospheric pressure. It is evident that the work area is the smallest possible when expansion is employed if the final volume is

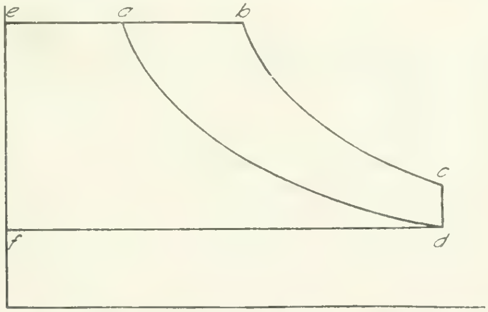

FIG. 3. WORK DIAGRAM FOR AIR. PARTIAL EXPANSION

not reduced below that attained when cold air is used. If the expansion of the reheated air is carried farther than shown, that is, if v_c is greater than v_d, the thermal efficiency will be somewhere between that of Case II and Case III.

In Fig. 3 the work resulting from the reheat supplied is represented by the area $a\ b\ c\ d$, which is the difference between areas $e\ b\ c\ d\ f$ and $c\ a\ d\ f$. Since

$$\text{area } e\ b\ c\ d\ f = \frac{n}{n-1}(p_a v_b - p_c v_c) + v_c (p_c - p_d)$$

$$\text{and area } c\ a\ d\ f = \frac{n}{n-1}(p_a v_a - p_d v_c)$$

$$\text{area } a\ b\ c\ d = \frac{n}{n-1}\left[(p_a v_b - p_c v_c) - (p_a v_a - p_d v_c)\right] +$$

$$v_c (p_c - p_d)$$

$$= \frac{n}{n-1} p_a (v_b - v_a) - \frac{1}{n-1} v_c (p_c - p_d) \text{ ft. lbs}$$

Substituting the appropriate values of $pv = BT$ in the preceding equation and multiplying by A,

$$\text{the area } a\ b\ c\ d = \frac{AB}{n-1}\left[n(T_b - T_a) - (T_c - T_d)\right] \text{ B.t.u}$$

The reheat supplied $= c_p (T_b - T_a)$; therefore the efficiency is

$$E = \frac{k-1}{k(n-1)}\left(n - \frac{T_c - T_d}{T_b - T_a}\right)$$

Since $$T_d = T_a \left(\frac{v_a}{v_d}\right)^{n-1}$$

$$T_b = T_a \left(\frac{v_b}{v_a}\right)$$

and $$T_c = T_b \left(\frac{v_b}{v_d}\right)^{n-1} = T_a \frac{v_b}{v_a}\left(\frac{v_b}{v_d}\right)^{n-1}$$

$$E = \frac{n\,(k-1)}{k\,(n-1)}\left[1 - \frac{1}{n}\left(\frac{v_a}{v_d}\right)^{n-1}\frac{\left(\frac{v_b}{v_a}\right)^{n} - 1}{\frac{v_b}{v_a} - 1}\right]$$

If adiabatic expansion is assumed, $n = k$ and the expression for efficiency reduces to

$$E = 1 - \frac{1}{k}\left(\frac{v_a}{v_d}\right)^{k-1}\frac{\left(\frac{v_b}{v_a}\right)^{k} - 1}{\frac{v_b}{v_a} - 1}$$

In this case the efficiency decreases as the volume v_b and temperature T_b increase.

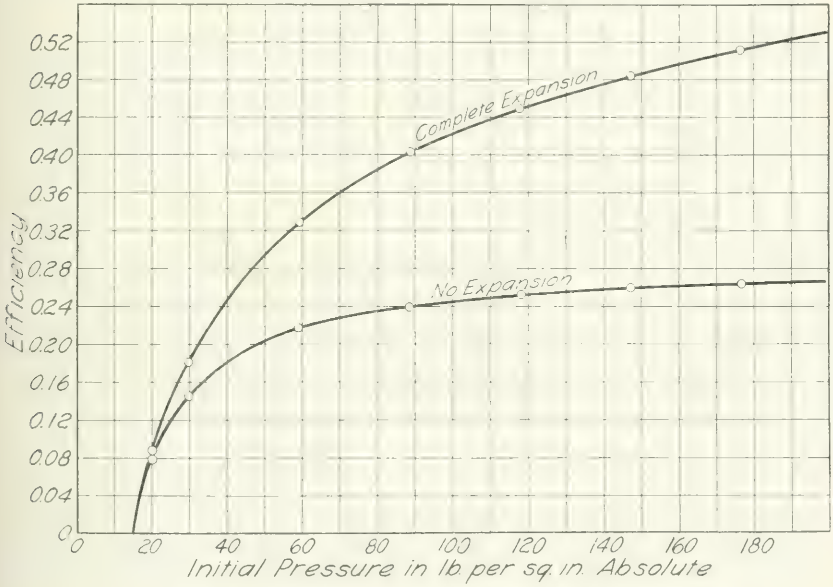

Fig. 4. Theoretical Efficiencies of Air Cycles

Reference to the efficiencies theoretically attainable in Cases I and II shows that they are independent of the reheat added and that they are functions of $\frac{p_d}{p_a}$ only. Fig. 4 shows the theoretical efficiencies for cycles having no expansion of the air and for those having complete adiabatic expansion.

III. Tests of Reheaters

7. *Method of Testing External Combustion Reheaters.*—Most of the tests of external combustion reheaters had a duration of eight hours, all conditions during that period being maintained as nearly constant as possible. Since the reheaters tested and the weight of fuel consumed were small, the percentage of error in fuel weights due to starting and stopping the tests would have been increased with a reduction in time. With eight-hour tests, however, it was felt that this error would be very small.

Before starting a test a preliminary run of about one hour was made to insure the maintenance of standard conditions from the beginning of the test. When the compressor, piping system, and reheater had reached thermal equilibrium, the fire was quickly drawn and rebuilt with weighed kindling. This process required about three minutes, during which the air was closed off from the reheater and the output of the compressor escaped through the relief valve on the air receivers. As soon as the fire was properly started the air was turned on and the test begun.

The fuel used, a coke of good quality, was fired in approximately equal weights, and at equal intervals of time during each test. The weights in different tests varied with the weight of air heated; and, in general, the interval of time was 15 to 20 minutes, except in the case of the first two or three firings where the interval was shorter. During the preliminary run the depth of the fire necessary for the assigned "load" was determined; then, at the beginning of the test, the fire was brought to this depth in the shortest possible time by using maximum draft, until the flue gas temperature was 100 to 150 degrees above the normal for the test. This building up period lasted 15 to 20 minutes and the second and third charges were fired as soon as the previous charge showed red on top.

The constancy of the temperature of the reheated air leaving the reheater was the regulative factor controlling the tests. To maintain a constant temperature, a controllable furnace draft was found indispensable. The draft in the stack, to which the reheater was connected, was induced and controlled by a jet of compressed air from a one-eighth inch pipe. A valve controlling the jet of air made it possible to obtain any draft from 0.05 to 0.40 inches of water. A thermometer inserted through a stuffing box directly in the stream of heated air made it possible to observe the temperature and to maintain it, with a variable draft, to within about 10 degrees Fahrenheit of the mean temperature.

A valve in the air pipe about ten feet from its entrance to the reheater was employed to maintain in the reheater a constant pressure of 100 to 110 pounds per square inch, gage, during most of the tests. The pressure in the receiver was 10 to 15 pounds higher. The flow of air was controlled by a second valve in the pipe leaving the reheater. The proper openings of these valves were determined by trial during the preliminary run, but slight readjustments were frequently necessary.

To assist in regulating the draft the temperature of the flue gases was determined by a Hoskins pyrometer. This temperature was kept as low as possible to permit the maintenance of the desired temperature of the reheated air. Samples of flue gas were taken in aspirator bottles over four-hour periods and analyzed during or immediately after the test by means of an Orsat apparatus filled with freshly prepared solutions.

After leaving the reheater the air passed through about 12 feet of uncovered pipe to a gage-box fitted with a standard orifice. A pressure of from 2 to 5 inches of water was maintained in the box and the discharge was computed by using the coefficients recommended by R. J. Durley.* As the relatively high temperatures might affect the coefficients of discharge, the Venturi tube, later used in metering the air for the engine tests, was connected in series with the reheater and gage-box, and several tests were run with air at different temperatures. As the ratio between the flow indicated by the meter and the gage-box followed the same law with and without the heater in operation, the use of Durley's coefficients for the measurement of air in the test was justified.

This same arrangement of apparatus was employed in calibrating the Venturi tube used in the tests. This tube is of a standard form manufactured by the Builders' Iron Foundry of Providence, Rhode Island. It is flanged for standard 2½-inch pipe connections and has a throat diameter of 0.75 inches. The standard differential mercury manometer supplied by the manufacturers of the tube was used in all tests. In certain air-steam mixture tests, made at a later date than those described in this bulletin, the question of the accuracy of the calibration of the Venturi tube was raised and various attempts were made to recalibrate the apparatus by methods other than the one originally employed. No one of these attempts was successful until use was made of the air weighing plant[*] used to calibrate instruments employed in the investigation of warm air furnaces. A careful recalibration of the Venturi tube indicated that, with Durley's coefficients, the original calibration against standard orifices gave weights of air which were 3 per cent low. Accordingly all computations involving weights of air originally made for this bulletin have been corrected to agree with this new calibration. This merely involved the multiplication of the weights of air as originally calculated by 1.03. The determination by means of the Venturi tube of the weight of air flowing involves much laborious computation. In order to reduce the labor and at the same time secure greater accuracy, large scale graphical charts, prepared for use with the Venturi tube, were employed in making the calculations. It seems unnecessary to present this material here.[†]

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* Durley, R. J. "On the Measurement of Air Flowing into the Atmosphere through Circular Orifices in Thin Plates and Under Small Differences of Pressure." Trans. A. S. M. E., v. 27, p. 193, 1906.

During the tests the following observations were recorded every 10 minutes:

(1) Temperature of air entering reheater
(2) Pressure of air entering reheater
(3) Temperature air leaving reheater
(4) Flue gas temperature
(5) Pressure in gage-box
(6) Temperature in gage-box
(7) Room temperature
(8) Draft

Other observations and records were:

(9) Barometric pressure (each hour)
(10) Diameter of orifice

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[*] For a description of the plant see "Investigation of Warm Air Furnaces and Heating Systems," Univ. of Ill. Eng. Exp. Sta., Bul. No. 120, 1921.

[†] The general theory of gaseous flow through Venturi tubes is presented in a paper, "The Flow of Fluids in a Venturi Tube," by Edgar P. Coleman. Trans. A. S. M. E., v. 28, p. 483.

(11) Position of damper

(12) Weight of kindling

(13) Time of each firing

(14) Net weight of fuel on scales after each firing

(15) Weight of fuel fired each firing

(16) Weight of residual fuel

(17) Weight of ash and refuse

(18) Flue gas analysis

Samples of each load of fuel placed on the scales were taken and sealed for analysis. Fig. 5 shows the general arrangement of testing apparatus, the figures referring to the numbered observations just listed. The fire was kept up to the standard condition until the end of the test. About two minutes before this time, the grates were thoroughly shaken and the ash and refuse collected, weighed, and preserved. Then at the end of the test the grates were dumped, and the fire quenched with the minimum quantity of water necessary. The residual fuel was then collected, weighed, and preserved.

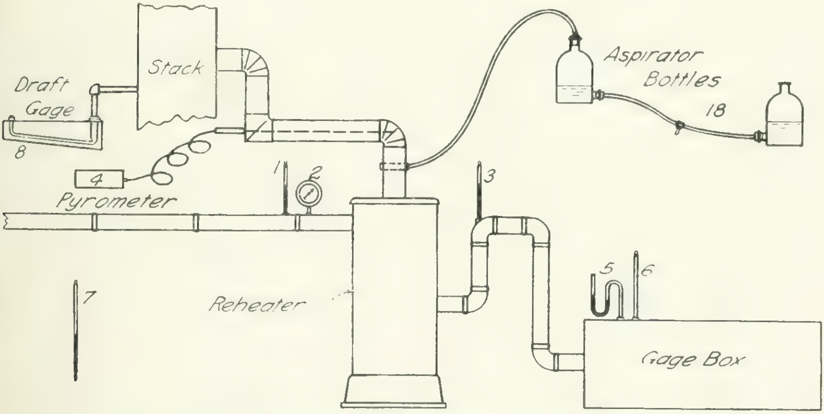

Fig. 5. General Arrangement of Apparatus for Testing External Combustion Reheater

8. *Definitions and Notation Employed.*—In rating compressed air equipment it is customary to express capacity in cubic feet of free air per minute. The term "free air" is loosely used, and without specific definition is meaningless. As employed in this bulletin the free air unit is one cubic foot of air at a temperature

of 62 degrees Fahrenheit and a pressure of 14.7 pounds per square inch absolute. The volume of a pound of air at 62 degrees Fahrenheit and 14.7 pounds per square inch absolute pressure is 13.147 cubic feet. The weight of a cubic foot of free air is 0.07608 pounds. One pound of air per hour is equivalent to 0.21906 cubic feet of free air per minute.

The specific heat of dry air at constant temperature, c_p, does not greatly vary within the temperature range employed in this investigation, but when moisture is present, the change of specific heat for small temperature changes is marked. According to Swann, the specific heat at constant pressure of dry air at 20 degrees centigrade (68 degrees Fahrenheit) is 0.24301. In computations made in this investigation the value of c_p for dry air was taken as 0.243, and it was assumed that the air before reheating was saturated with water vapor. By the theory of mixture of gases and vapors it is possible to calculate the value of c_p for such mixtures. Values of c_p thus calculated are given in the various tables presented.

The efficiency of a reheater is the ratio of the heat absorbed by the compressed air to the calorific value of the fuel charged against the reheater. The numerator of this fraction is the product of the weight of air heated in pounds, the specific heat of the air at constant pressure, and the rise in temperature in degrees Fahrenheit; the denominator is the calorific value of the total fuel fired minus the calorific value of the residual fuel either with or without the fuel in the ash—depending upon whether the efficiency of the grate is excluded or included. In internal combustion reheaters using liquid fuel the denominator is the calorific value of the liquid fuel consumed.

For convenience of reference the designations of all tests are explained as follows:

Reheater tests are designated by two numbers and a letter. The first number denotes the approximate quantity of free air in cubic feet per minute; the second, the approximate temperature of the air after reheating; and the letter is added only for the sake of distinction; e.g., 150–400–a.

Engine tests using air are designated by four numbers: first, a serial number; second, the gage pressure at the throttle; third, the temperature at the throttle; and fourth, the approximate cut-off in per cent of the engine stroke; e.g., 50–100–200–25.

Air-steam engine tests are denoted by the letters A–S followed by the gage pressure at the throttle and the approximate percentage of steam in the mixture; e.g., A–S–100–10.

Tests with steam alone are designated by the letter S, followed by the gage pressure at the throttle; e.g., S–80.

9. *Tests of Type I Reheater.*—This reheater consists of a cast iron fire-pot in the form of the frustrum of a cone, with base downward, and surrounded by a shell with an annular space through which the air circulates. A detachable grate at the bottom of the fire-pot made it possible to draw the fire in a few seconds. Since the air heating surface completely surrounds the fire-pot, the temperature of the exterior of the reheater is not greater than that of the reheated air; hence the loss by radiation is small. Type I reheater has an effective heating surface of 2.9 square feet, and, when reheating air to 400 degrees Fahrenheit, a capacity of about 200 cubic feet of free air per minute.

Coke was used as a fuel in all external combustion reheater tests. The samples of coke, residual fuel, ash, and refuse collected during the several tests of the Type I reheater were analyzed by the Department of Chemistry of the University of Illinois with the following results:

	B.t.u.
(1) Calorific value per pound of air-dry coke as fired, by the oxygen calorimeter	12 992

(2) Proximate analysis of air-dry coke:

	Per Cent
Fixed carbon	88.76
Volatile matter	1.65
Moisture	0.55
Ash	9.04
	100.00
Moisture on drying	0.00
Sulphur determined separately	0.25

(3) Ultimate analysis of air-dry coke:

Carbon	87.21
Hydrogen	0.74
Oxygen	1.34
Nitrogen	0.87
Sulphur	0.25
Ash	9.04
Moisture at 105 deg. C.	0.55
	100.00

(4) Proximate analysis of residual fuel:

Carbon	86.30
Earthy matter	13.40
Moisture	0.30
	100.00

(5) Proximate analysis of ash and refuse:

Carbon	30.92
Earthy matter	68.93
Moisture	0.15
	100.00

It was assumed that each pound of wood used as kindling had a calorific value of 0.4 pound of coke, and that each pound of oil used in kindling the fire had a calorific value of 1.47 pound of coke. The calorific value of carbon is taken as 14 540 B.t.u. per pound. From these assumptions and the data from the several analyses the various items presented in Table 1 were computed. In test 100 300 b the weight of ash and refuse was not recorded. Since they were obtained by computing the ash and refuse from the average grate efficiency, items 18 and 20 for this test are enclosed in brackets to indicate that their accuracy is open to some question.

Table 2 presents various data and the economic results of the tests of Type I reheater. The several numbered columns of the table present average recorded data or computed results. The methods of computing the various items are either indicated or obvious without further explanation. The efficiencies shown under items 14 and 15 are of chief interest. The efficiencies, exclusive of grate, vary from 48.0 to 61.5 per cent. The best efficiencies were obtained when from 100 to 150 cubic feet of free air per minute were reheated.

10. *Tests of Type II Reheater.*—In the Type II reheater the compressed air flowed through a series of cast iron coils set in a casing so that the coils were 12 to 15 inches above the normal level of the fire. No baffling was provided to force the heated gas over the coils, and the area of the furnace and casing exposed to the outer air was large; consequently the loss of heat by radiation was great. The heating surface of the coils was 18.6 square feet. The tests of this reheater were made in exactly the same manner as has been described for the Type I reheater.

TABLE 1

RESULTS OF TESTS OF TYPE I REHEATER

Designation			Gage-box Data					Gross Fuel per Test		
	1	2	3	4	5	6	7	8	9	10
	Date	Length of Test (hrs.)	Diameter of Orifice (in.)	Pressure in Gage-box (in. Water)	Temperature in Gage-box (deg. F.)	Barometer (in. Mercury)	Gage-box Coefficient (Durley's Curves)	Coke (lb. per Test)	Wood (lb. per Test)	Oil (lb. per Test)
50–400–a	6–10–12	8.00	1.5	3.56	241	29.5	0.603	24.0	3.00	0.30
100–400–a	6–11–12	8.00	2.0	4.86	297	29.4	0.601	42.5	3.00	0.25
150–400–a	6–12–12	8.00	2.5	4.59	316	29.2	0.598	52.0	3.00	0.25
200–400–a	6–14–12	8.00	3.5	2.04	308	28.8	0.597	70.0	4.00	0.30
275–200–b	5–30–12	6.33	3.5	3.88	203	29.3	0.595	39.5	3.00	0.66
275–200–c	6– 1–12	8.50	3.5	4.02	223	29.1	0.595	50.0	3.00	0.60
100–250–b	6– 3–12	8.00	2.5	2.15	220	29.2	0.599	23.0	3.00	0.40
200–300–b	6–13–12	4.00	3.5	2.16	287	29.1	0.597	33.0	3.00	0.30
100–300–b	5–27–12	5.00	2.5	2.08	225	28.8	0.599	20.0	1.25	0.10
100–300–c	5–29–12	8.00	2.5	2.06	243	29.1	0.599	37.5	3.00	0.30
100–400–b	6– 5–12	8.00	2.5	2.10	291	29.3	0.599	44.0	3.00	0.30

TABLE 1 (CONTINUED)

RESULTS OF TESTS OF TYPE I REHEATER

Designation	Fuel Equivalents						Net Fuel			
	11	12	13	14	15	16	17	18	19	20
	Ash and Refuse (lb. per Test)	Residual Fuel (lb. per Test)	Coke Equivalent of Wood and Oil (lb. per Test)	Total Equivalent Coke (lb. per Test) (Item 8 + Item 13)	Coke Equivalent of Ash and Refuse (lb. per Test)	Coke Equivalent of Residual Fuel (lb. per Test)	Net Equivalent Coke Charged against Reheater (lb. per Test) (Item 14 − Item 16)	Net Equivalent Coke Consumed (lb. per Test) (Item 17 − Item 15)	Net Equivalent Coke Charged against Reheater (lb. per hr.) (Item 17 ÷ Item 2)	Net Equivalent Coke Consumed (lb. per Test) (Item 18 ÷ Item 2)
50-400-a	1.50	4.50	1.64	25.64	0.52	4.36	21.28	20.76	2.66	2.60
100-400-a	2.50	11.00	1.56	44.06	0.87	10.70	33.36	32.49	4.17	4.06
150-400-a	4.50	8.25	1.56	53.56	1.56	7.94	45.62	44.06	5.71	5.51
200-400-a	7.75	12.00	2.03	72.03	2.71	11.66	60.37	57.66	7.55	7.21
275-200-b	4.42	13.11	2.16	41.66	1.54	12.72	28.94	27.40	1.56	4.33
275-200-c	4.50	9.75	2.07	52.07	1.56	9.49	42.58	41.02	5.02	4.82
100-250-b	1.75	3.00	1.78	24.78	0.61	2.92	21.86	21.25	2.74	2.66
200-300-b	2.00	9.50	1.64	34.64	0.70	9.23	25.41	24.71	6.35	6.18
100-300-b	....	7.88	0.65	20.65	..	7.66	13.00	[12.55]	2.60	[2.51]
100-300-c	2.71	11.23	1.64	39.14	0.94	10.93	28.21	27.27	3.53	3.41
100-400-b	2.50	11.00	1.64	45.64	0.87	10.70	34.94	34.67	4.37	4.26

TABLE 2

RESULTS OF TESTS OF TYPE I REHEATER

	1	2	3	4	5	6	7	8	9	10	11	12
Designation	Length of Test (hr.)	Pressure of Air Entering Reheater (lb. per sq. in. Gage)	Temperature Air Entering Reheater (deg. F.)	Temperature Air Leaving Reheater (deg. F.)	Rise of Temperature	Air Reheated (lb. per hr.)	Air Reheated in cu. ft. of Free Air per min. (62° F., 14.7 lb. Abs.)	Specific Heat at Const. Press. Air Saturated Entering Reheater	B. t. u. Transmitted to Compressed Air (per hr.)	B. t. u. Transmitted per hr. per sq. ft. of Heating Surface	Total B. t. u. Supplied per hr.	Grate Loss per hr. in B. t. u
50-400 a	8.00	109.7	98.1	402.8	304.7	218	47.7	0.244	16170	5590	34550	844
100-400 a	8.00	109.9	111.5	409.2	297.7	432	95.0	0.244	31440	10860	54250	1406
150-400 a	8.00	109.6	121.0	397.2	276.2	644	141.0	0.245	43750	15110	73665	2530
200-400-a	8.00	117.4	132.7	385.9	252.3	838	183.5	0.245	51810	17890	98112	1363
275-200 b	6.33	99.4	128.1	225.0	96.9	1253	275.0	0.246	29810	10290	58675	2388
275-200-c	8.50	100.1	139.5	247.3	107.8	1232	271.0	0.246	33130	11420	67530	2380
100-250 b	8.00	108.0	111.2	278.7	167.5	471	103.0	0.244	19250	6640	35545	984
200-300-b	4.00	108.5	126.3	343.8	217.2	880	192.5	0.245	46750	16140	82627	2250
100-300 b	5.00	95.7	108.8	287.0	178.2	459	100.0	0.244	19960	6880	33820	1210
100-300 c	8.00	101.4	101.6	324.4	222.8	454	99.4	0.244	24620	8520	45896	1525
100-400 b	8.00	106.9	110.1	401.5	291.4	444	97.5	0.244	31570	10870	56805	1406

TABLE 2 (CONTINUED)

RESULTS OF TESTS OF TYPE I REHEATER

	13	14	15	16	17	18	19	20	21	22		
Designation	B. t. u Liberated per hr. by Combustion	Efficiency Including Grate (Per cent)	Efficiency Excluding Grate (Per cent)	Net Equivalent Coke Charged Against Reheater (lb. per hr.)	Pounds Coke Required to Raise 1000 lb. Air 100° F. (Item 16) ÷ $\left(\frac{\text{It.5}}{100} \times \frac{\text{It.6}}{1000}\right)$	Draft At Base of Stack in in. of Water	Room Temperature (deg. F.)	Barometer (in. Mercury)	Temperature of Flue Gases (deg. F.)	Flue Gas Analysis		
										Carbon Dioxide (CO_2)	Oxygen (O_2)	Carbon Monoxide (CO)
50–400–a	33715	46.8	48.0	2.66	4.00	0.09	80	29.5	568	8.9	10.8	0.0
100–400–a	52844	57.9	59.4	4.17	3.24	0.15	84	29.4	611	9.1	10.1	0.0
150–400–a	71135	59.4	61.5	5.71	3.20	0.24	87	29.2	581	10.8	8.1	0.0
200–400–a	93749	52.8	55.2	7.55	3.57	0.35	84	28.8	601	10.5	8.4	0.0
275–200–b	56287	50.6	53.0	4.56	3.75	0.04	75	29.3	572	12.8	5.7	0.0
275–200–c	65150	49.1	50.9	5.02	3.77	0.03	85	29.1	561	13.2	5.0	0.0
100–250–b	34561	54.4	55.6	2.74	3.47	0.08	83	29.2	606	6.5	12.7	0.5
200–300–b	80377	56.6	58.1	6.35	3.32	0.30	80	29.1	682	9.5	9.5	0.0
100–300–b	[32610]	59.0	[61.1]	2.60	3.18	0.05	86	28.8	475	....	....	....
100–300–c	44371	52.4	55.6	3.53	3.50	0.05	73	29.1	607	11.3	7.7	0.0
100–400–b	55399	55.6	57.0	4.37	3.38	0.14	81	29.3	629	12.7	6.8	0.5

Analyses of the fuel, residual fuel, ash, and refuse gave the following results:

	B.t.u.
(1) Calorific value per pound of air-dry coke as fired, by the oxygen calorimeter	12 718

(2) Proximate analysis of air-dry coke:

	Per Cent
Fixed Carbon	87.25
Volatile matter	2.24
Ash	0.50
Moisture	10.01
	100.00
Moisture on drying	0.00
Sulphur separately determined	0.66

(3) Ultimate analysis of coke:

Carbon	83.95
Hydrogen	1.20
Oxygen	2.68
Nitrogen	1.00
Sulphur	0.66
Ash	10.01
Moisture at 105 deg. C.	0.50
	100.00

(4) Proximate analysis of residual fuel:

Carbon	58.64
Earthy matter	41.22
Moisture	0.14
	100.00

(5) Proximate analysis of ash and refuse:

Carbon	18.62
Earthy matter	81.38
Moisture	0.00
	100.00

Tables 3 and 4 present all the essential data and results of the tests of this reheater. The efficiencies obtained were low, due to excessive loss of heat by radiation and in the flue gases. With small loads the fire was necessarily thin and the excess air large; with larger loads the loss from excess air was reduced, although the flue gas temperature was abnormally high, and the loss of heat in the

TABLE 3

RESULTS OF TESTS OF TYPE II REHEATER

				Gage-box Data				Gross Fuel per Test		
	1	2	3	4	5	6	7	8	9	10
Designation	Date	Length of Test (hrs.)	Diameter of Orifice (in.)	Pressure in Gage-box (in. Water)	Temper-ature in Gage-box (deg. F.)	Barometer (in. Mercury)	Gage-box Coefficient (Durley's Curves)	Coke (lb. per Test)	Wood (lb. per Test)	Oil (lb. per Test)
50–400-d	6–22–12	8.00	1.5	3.58	242	29.5	0.602	65.0	10.00	0.25
100–400-d	6–24–12	8.00	2.0	4.80	301	29.3	0.600	100.0	6.00	0.75
150–400-d	10–23–12	8.00	3.0	2.19	311	29.4	0.598	103.5	8.60	1.50
200–400-d	10–22–12	8.00	3.0	3.84	308	29.2	0.597	133.5	9.50	1.30
250–400-d	10–24–12	8.00	3.5	3.27	325	29.4	0.597	145.5	8.75	1.50
300–400-d	10–25–12	8.00	3.5	4.75	341	29.3	0.595	174.5	9.50	1.50
350–400-d	10–26–12	8.00	4.0	3.96	348	29.3	0.595	189.5	9.50	1.50

TABLE 3 (CONTINUED

RESULTS OF TESTS OF TYPE II REHEATER

Designation	Fuel Equivalents						Net Fuel			
	11	12	13	14	15	16	17	18	19	20
	Ash and Refuse (lb. per Test)	Residual Fuel (lb. per Test)	Coke Equivalent of Wood and Oil (lb. per Test)	Total Equivalent Coke (lb. per Test) (Item 8 + Item 13)	Coke Equivalent of Ash and Refuse (lb. per Test)	Coke Equivalent of Residual Fuel (lb. per Test)	Net Equivalent Coke Charged against Reheater (lb. per Test) (Item 14 — Item 16)	Net Equivalent Coke Consumed (lb. per Test) (Item 17 — Item 15)	Net Equivalent Coke Charged against Reheater (lb. per hr.) (Item 17 ÷ Item 2)	Net Equivalent Coke Consumed (lb. per hr.) (Item 18 ÷ Item 2)
50-400-d	4.75	10.00	4.36	69.36	1.01	6.71	62.65	61.64	7.83	7.71
100-400-d	6.40	14.70	3.49	103.49	1.35	9.88	93.61	92.26	11.70	11.53
150-400-d	6.50	8.50	5.63	109.12	1.38	5.70	103.42	102.04	12.93	12.76
200-400-d	8.00	11.00	5.69	139.19	1.70	7.39	131.80	130.10	16.48	16.26
250-400-d	6.50	14.00	5.68	151.18	1.38	9.41	141.77	140.39	17.72	17.55
300-400-d	8.00	21.50	5.98	180.48	1.70	14.42	166.06	164.36	20.76	20.55
350-400-d	10.00	15.00	5.98	195.48	2.12	10.03	185.45	183.33	23.18	22.92

TABLE 4

RESULTS OF TESTS OF TYPE II REHEATER

	1	2	3	4	5	6	7	8	9	10	11	12
Designation	Length of Test (hr.)	Pressure of Air Entering Reheater (lb. per sq.) in. Gage)	Temperature Air Entering Reheater (deg. F.)	Temperature Air Leaving Reheater (deg. F.)	Rise of Temperature	Air Reheated (lb. per hr.)	Air Reheated in cu. ft. of Free Air per min. (62° F., 14.7 lb. Abs.)	Specific Heat at Const. Press. Air Saturated Entering Reheater	B. t. u. Transmitted to Compressed Air (per hr.)	B. t. u. Transmitted per hr. per sq. ft. of Heating Surface	Total B. t. u. Supplied per hr.	Grate Loss per hr. in B. t. u.
50-400-d	8.00	82.3	98.3	390.6	292.3	232	51	0.242	16397	882	99602	1606
100-400-d	8.00	82.8	105.4	406.0	300.6	428	94	0.243	31286	1682	148860	2166
150-400-d	8.00	104.4	109.1	405.6	296.5	661	145	0.243	47640	2561	164425	2200
200-400-d	8.00	102.3	118.8	380.2	261.4	868	190	0.245	55604	2989	209556	2716
250-400-d	8.00	106.6	122.2	392.4	270.2	1078	236	0.245	71353	3837	225425	2200
300-400-d	8.00	106.7	136.1	399.5	263.4	1279	280	0.246	82844	4454	263994	2766
350-400-d	8.00	106.5	146.4	398.0	251.6	1513	333	0.247	94060	5058	294775	3387

TABLE 4 (CONTINUED)

RESULTS OF TESTS OF TYPE II REHEATER

	13	14	15	16	17	18	19	20	21	22		
Designation	B. t. u. Liberated per hr. by Combustion	Efficiency Including Grate (Per cent)	Efficiency Excluding Grate (Per cent)	Net Equivalent Coke Charged Against Reheater (lb. per hr.)	Pounds Coke Required to Raise 1000 lb. Air 100° F. (Item 16 ÷ $\left[\frac{It.5}{100} \times \frac{It.6}{1000}\right]$	Draft At Base of Stack (in. Water)	Room Temperature (deg. F.)	Barometer (in. Mercury)	Temperature of Flue Gases (deg. F.)	Flue Gas Analysis		
										Carbon Dioxide (CO_2)	Oxygen (O_2)	Carbon Monoxide (CO)
50-400-d	97996	16.5	16.7	7.83	11.50	0.14	89	29.5	415	4.6	16.3	0
100-400-d	146694	21.0	21.3	11.70	9.10	0.14	100	29.3	495	6.0	14.0	0
150-400-d	162225	29.0	29.4	12.93	6.60	0.18	67	29.4	574	5.6	14.2	0
200-400-d	206850	26.5	26.9	16.48	7.27	0.35	73	29.2	599	4.5	15.5	0
250-400-d	223225	31.7	32.0	17.72	6.08	0.21	70	29.4	691	8.3	11.6	0
300-400-d	261288	31.4	31.7	20.76	6.15	0.20	76	29.3	735	9.7	10.6	0
350-400-d	291388	31.9	32.3	23.18	6.09	0.24	75	29.3	777	11.4	8.8	0

flue gas large. While the efficiencies might have been somewhat improved by substituting a long flaming coal for coke, the general design of the reheater is not conducive to fuel economy.

11. *Tests of Type III Reheater.*—This reheater is of the internal combustion type using gasoline as fuel. Its capacity is about 250 cubic feet of free air per minute, heated to a temperature of 400 degrees Fahrenheit. The sectional view of the reheater, Fig. 6, shows the method of operation. Cold air enters the reheater at *A* and the reheated air leaves at *E*. The incoming air enters the space *B* and its pressure is exerted on the surface of the gasoline, *I*, in the base of the reheater. *C* is an adjustable spring loaded valve set to cause a slight reduction of pressure in the air, which flows through the

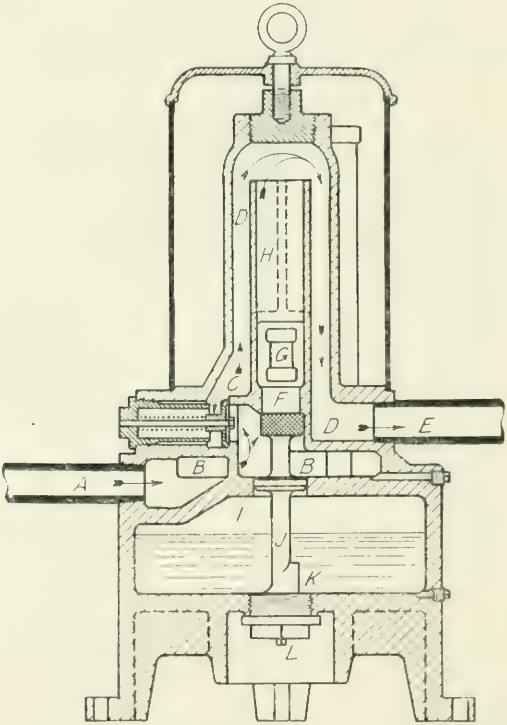

Fig. 6. Sectional View of Internal Combustion Reheater

valve into the space *D* and out at *E*. A tube, *J*, having an opening, *K*, from the gasoline reservoir leads to a needle valve and atomizer at *F*, adjustable at *L*; above the atomizer a combustion tube opens directly into the space *D*. Since the air pressure on the surface of the gasoline is slightly above that in space *D* and tube *H*, gasoline is forced through the atomizing tube upon the electrical ignition coil, *G*. The gasoline mixture burns in *H* and the products of combustion pass out with the reheated air.

In actual operation it was found difficult to secure and maintain the proper adjustment of the differential valve, *C*, and the atomizer. Considerable difficulty with the electrical ignition device was also experienced. Despite these difficulties, however, this reheater was used throughout the various engine tests described in a later section, and in general it proved satisfactory and convenient. It is decidedly easier to maintain constancy of reheat with this reheater than with the external combustion type. The slightest change in the position of the needle valve controlling the flow of gasoline affected at once the temperature of the out-going air.

The tests of this reheater were relatively simple. The Venturi tube already described was employed to measure the air. Its arrangement is shown in Fig. 7, and that of the reheater in Fig. 8.

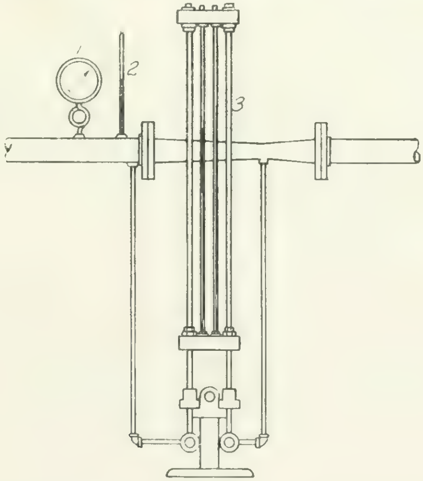

FIG. 7. APPARATUS FOR TESTING INTERNAL COMBUSTION REHEATER; GENERAL ARRANGEMENT OF VENTURI TUBE

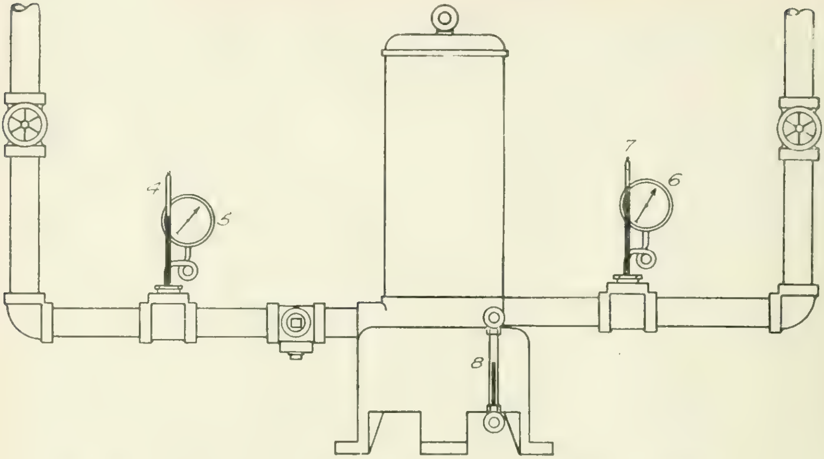

FIG. 8. APPARATUS FOR TESTING INTERNAL COMBUSTION REHEATER; GENERAL ARRANGEMENT OF REHEATER

The numbers shown in these two figures refer to the various observations necessary, as follows:

(1) Upstream pressure on tube (every 10 minutes)
(2) Upstream temperature (every 10 minutes)
(3) Manometer (every 2 minutes)
(4) Temperature of air entering reheater (every 10 minutes)
(5) Pressure of air entering reheater (every 10 minutes)
(6) Pressure of air leaving reheater (every 10 minutes)
(7) Temperature of air leaving reheater (every 5 minutes)
(8) Gasoline gage at beginning and end of test
(9) Room temperature (once for each test)
(10) Barometer (once for each test)

With three exceptions the duration of each test was one hour; two tests were continued for 1½ hours, and one—an overload test designated as 370–400–f—was stopped after 30 minutes because of the excessive load on the compressor. The gasoline used in the tests of this reheater had a specific gravity of 0.732 and a calorific value of 20 309 B.t.u. per pound.

Fig. 9 shows the drop of pressure through the reheater at different rates of air flow. During test 370–400–f, the compressor was unable to deliver the air at the normal pressure of 100 pounds gage;

in fact the average pressure was only 48 pounds gage. This probably had an effect on the loss of pressure with an air flow of 370 cubic feet of free air per minute.

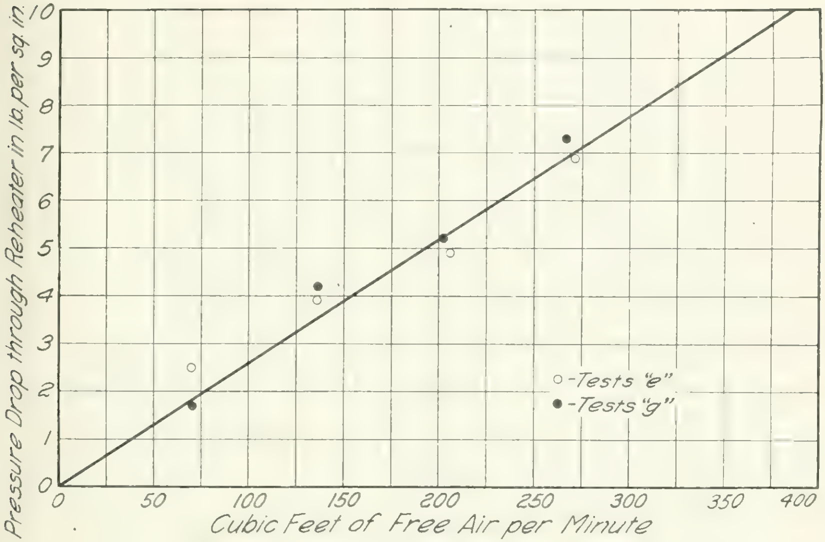

FIG. 9. DROP OF PRESSURE IN REHEATER

Table 5 presents the various observed quantities and the computed results. Item 17, the efficiency of the reheater, shows that due to more perfect vaporization and combustion of the gasoline the efficiency is better for the larger loads and higher temperatures of the reheated air. During some of the engine tests described in a later section, the temperature of the reheated air was maintained at 200 degrees Fahrenheit, and in some instances with comparatively small rates of flow. Under these conditions the combustion was so poor that there was a strong odor of unburned gasoline in the engine exhaust. Since the loss of heat by radiation could not have been large during any of the tests, it is evident that under the most favorable conditions maintained the loss from incomplete combustion was excessive.

TABLE 5

RESULTS OF TESTS OF TYPE III REHEATER

	1	2	3	4	5	6	7	8	9	
Designation	Length of Test (hr.)	Pressure of Air Entering Reheater (lb per sq. in. Gage)	Pressure of Air Leaving Reheater (lb. per sq. in. Gage)	Pressure Drop Through Reheater	Temperature Air Entering Reheater (deg. F.)	Temperature Air Leaving Reheater (deg. F.)	Rise of Temperature	Specific Heat at Const. Press. Air Saturated Entering Reheater	Venturi Meter Data	
									Temperature Upstream (deg. F.)	Pressure Upstream (lb. per sq. in. Gage
70-400 e	1.5	99.3	96.8	2.5	99.0	402.0	363.0	0.244	112	99.4
140-400 e	1.0	99.5	95.6	3.9	112.3	403.0	290.7	0.245	132	99.7
200-400 e	1.0	99.7	94.8	4.9	125.3	400.0	274.7	0.247	154	100.0
260-400 e	1.5	100.8	93.9	6.9	140.0	405.0	265.0	0.250	168	101.8
370-400 f	0.5	48.0	34.5	13.5	156.0	404.0	248.0	0.264	180	52.8
70-300 g	1.0	99.1	97.4	1.7	110.0	300.2	190.2	0.244	122	100.4
140-300 g	1.0	100.5	96.3	4.2	113.1	302.1	189.0	0.245	133	100.5
200-300 g	1.0	100.0	94.8	5.2	128.4	302.2	173.8	0.247	148	100.0
260-300 g	1.0	99.3	92.0	7.3	137.4	308.2	170.8	0.250	165	99.4

TABLE 5 (CONTINUED)

RESULTS OF TESTS OF TYPE III REHEATER

	9		10	11	12	13	14	15	16	17	18
Designation	Venturi Meter Data										
	Manometer (in. Mercury)	Barometer (in. Mercury)	Air Reheated (lb. per hr.)	Air Reheated (cu. ft. of Free Air per min.) (62° F., 14.7 lb. Abs.)	B. t. u. Absorbed by Compressed Air (per hr.)	Fuel Used (lb. per hr.)	Lb. of Gasoline Required to Raise 1000 lb. of Air 100° F.*	Coke Equivalent of Item 14 (calorific Value 13 000 B. t. u. per lb.)	B. t. u. in Fuel Used (per hr.)	Efficiency (Per cent)	Room Temp. (deg. F.)
70-400-e	0.35	29.2	328	69.7	24215	1.57	1.58	2.46	31884	75.9	92
140-400-e	1.32	29.2	624	137.0	44455	2.70	1.49	2.33	54834	81.1	97
200-400-e	3.21	29.2	940	206.0	63807	3.72	1.14	2.25	75549	84.5	99
260-400-e	5.91	29.3	1240	271.9	82159	4.73	1.44	2.25	96062	85.5	98
370-400-f	26.72	29.3	1727	378.0	113020	6.80	1.58	2.46	138100	81.9	98
70-300-g	0.35	29.1	326	71.4	15105	1.04	1.68	2.62	21121	71.5	83
140-300-g	1.30	29.1	621	136.2	28760	1.77	1.50	2.35	35947	80.0	83
200-300-g	3.05	29.1	920	201.8	39532	2.55	1.59	2.49	51788	76.3	83
260-300-g	5.81	29.1	1214	266.0	51853	3.47	1.67	2.61	70492	73.5	83

*(Item 13) ÷ $\left(\frac{\text{Item 7}}{100}\right) \times \left(\frac{\text{Item 10}}{1000}\right)$

IV. Tests of Engine Using Air as Working Fluid

12. *Purposes of Tests.*—A series of tests, during which air at various initial temperatures was used expansively as the working fluid in a small Corliss engine, was made to determine first, what proportion of the heat applied in the reheating process is utilized in the development of indicated work within the cylinder, and as effective or "brake" work delivered by the engine; and second, the efficiencies of the actual engine using reheated air in comparison with those theoretically obtainable. To secure the desired results it was necessary to establish the efficiencies of the engine under various conditions of operation without reheat, and to compare these results with those obtained under similar conditions of operation with the air heated to the several temperatures, specified by the range of the investigation. Since the injection of steam in the air pipe has occasionally been employed as a means of reheating the air in order to overcome operating difficulties with the motor, the experimental investigation was extended to determine the effectiveness of air-steam mixtures in comparison with steam and unreheated air used separately.

13. *Description of Experimental Plant.*—Air for the tests was supplied by an Ingersoll-Sergeant, two-stage, cross compound compressor, having an approximate capacity of 375 cubic feet of free air per minute compressed to 100 pounds per square inch gage. From the compressor the air was delivered to a receiver having a capacity of 65 cubic feet. From there it passed through the Venturi tube previously described, to a second receiver—capacity, 17 cubic feet—employed to reduce the pulsations of pressure in the Venturi tube, and from thence through the internal combustion reheater, described in Section 11, to the engine. During the tests using air only the exhaust air from the engine was allowed to escape into the laboratory through a short exhaust pipe; during the tests of air-steam mixtures the exhaust pipe was connected to a Dean surface condenser having a tube surface of 100 square feet. Above the engine throttle valve the air line was provided with an effective mixing chamber and water separator for use when steam was injected into the air.

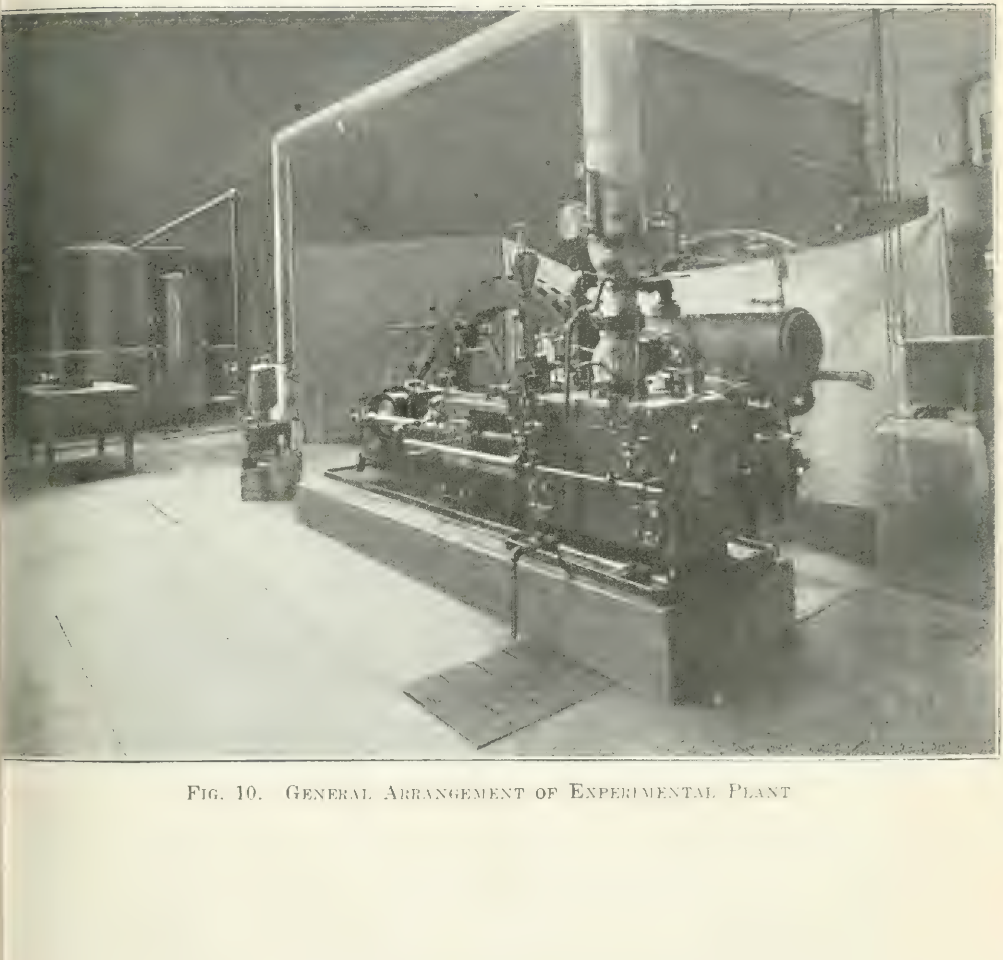

FIG. 10. GENERAL ARRANGEMENT OF EXPERIMENTAL PLANT

The engine used in this investigation was of the simple Corliss type built by the Murray Iron Works of Burlington, Iowa. Its principal dimensions are as follows:

	Inches
Diameter of Cylinder	8.00
Length of Stroke	18.05
Diameter of Piston Rod	1.87
Diameter of Steam Pipe	2.50
Diameter of Exhaust Pipe	3.00

Cylinder clearance in per cent of the total piston displacement is:

	Per Cent
For the Head End of the Cylinder	5.48
For the Crank End of the Cylinder	5.63

A specially designed indicator reducing motion of great accuracy was permanently attached to the engine, which was also equipped with a positive revolution counter and a Prony brake of excellent design and unusual steadiness. A pair of Tabor outside spring indicators were used in all tests.

Wherever necessary, pressures were determined by carefully calibrated standard test gages graduated in one pound increments. Temperatures were determined by standardized thermometers, those at the throttle and in the steam chest being secured by thermometers immersed through stuffing boxes directly in the stream of air. The temperature of the exhaust was secured by a pentane thermometer graduated in degrees centigrade, and placed in a mercury well in the exhaust pipe as close to the cylinder as possible. These temperatures are not an accurate measure of the actual temperatures in the cylinder during the exhaust stroke, since heat was rapidly absorbed from the surrounding atmosphere by the exhaust air in the pipe. During the earlier tests a hand pump was employed for lubricating the cylinder; in the later, a forced feed lubricator was attached to the engine. Fig. 10 shows the general arrangement of the plant.

14. *Method of Conducting Test.*—To insure a condition of thermal equilibrium in the piping and engine, the plant was operated for a period of one and one-half to two hours at the load to be maintained during the test. The cylinder of the engine was slow in reaching uniform temperature conditions, particularly when the point of cut-off was less than one-quarter of the stroke. When the difference between the temperatures at the throttle valve and in the steam chest

became nearly constant it was possible to begin the actual test. As soon as thermal equilibrium was obtained, the predetermined conditions of pressure, air temperature, cut-off, etc., to be maintained during the test were established and the test was started. The order in which tests were undertaken was so chosen as to require the least possible change in conditions between tests, for any change involved the lapse of considerable time between tests to re-establish thermal equilibrium. The tests using air were continued for thirty minutes, during which the following measurements were secured:

(1) Upstream temperature of air (every 10 minutes)
(2) Temperature entering reheater (every 10 minutes)
(3) Temperature leaving reheater (every 10 minutes
(4) Temperature in steam chest (every 10 minutes)
(5) Temperature of exhaust (every 10 minutes)
(6) Temperature of room (every 10 minutes)
(7) Upstream pressure (every 10 minutes)
(8) Manometer reading of Venturi meter (every 2 minutes)
(9) Indicator diagrams (every 5 minutes)
(10) Height of barometer (once for each test)
(11) Height of gasoline in reservoir of reheater at beginning and end of test

The pressure and temperature of the air at the throttle and the load on the brake were adjusted at the beginning of the test and maintained constant during the test.

When the reheater was not in use the air reached the engine from the compressor at about 100 degrees Fahrenheit; therefore, this was established as the basic temperature of the unreheated air. It was originally intended to cool the air in the receiver to room temperature by a spray of water, but this plan was abandoned when it was found that with air at 100 degrees Fahrenheit, great trouble was experienced through the formation of ice in the exhaust ports and pipe.

15. *Classification of Air Tests.*—For convenience the meter, indicator, brake horse power, and reheater data for all the tests in which air alone was used are given in Table 6. Of the 63 tests reported those numbered 1 to 16, inclusive, are tests in which the expansion of the air was complete to atmospheric pressure; those numbered 17 to 22, inclusive, are the first tests with variable cut-off; those numbered

Table 6

Observed Values for Engine Tests with Reheated Air

		Venturi Meter				Indicator Data				b. h. p. Data		Reheater		
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Designation	Date	Temperature of Air Upstream (deg. F.)	Pressure of Air Upstream (lb. per sq. in. Gage)	Barometer (in. Mercury)	Venturi Meter Manometer (in. Mercury)	m. e. p. Head End of Cylinder	m. e. p. Crank End of Cylinder	i. h. p. Head End of Cylinder	i. h. p. Crank End of Cylinder	Revolutions per min.	Weight on Scales	Temperature Entering Reheater (deg. F.)	Temperature Leaving Reheater (deg. F.)	Rise of Temperature (deg. F.)
1-100-400-15	2-19-13	132	103	29.0	0.83	28.0	29.8	6.80	6.86	106.1	116	104	452	348
2-100-300-15	2-19-13	135	103	29.0	0.91	27.7	28.9	6.73	6.65	106.1	111	103	328	225
3-100-200-15	2-19-13	115	103	29.0	1.04	27.6	29.7	6.71	6.83	106.1	106	97	215	118
4-100-100-15	2-20-13	130	103	29.2	1.54	28.6	31.2	6.91	7.17	105.9	101	....	....	...
5-80-400-20	2-24-13	101	82	29.4	0.87	26.5	27.4	6.40	6.26	105.4	106	80	446	366
6-80-300-20	2-24-13	90	82	29.4	1.00	26.3	26.9	6.35	6.16	105.5	101	74	338	264
7-80-200-20	2-21-13	126	83	29.0	1.07	25.9	26.5	6.27	6.08	105.8	86	92	227	135
8-80-100-20	2-20-13	116	83	29.2	1.80	27.7	28.3	6.68	6.47	105.3	90	...		
9-60-400-25	2-24-13	94	62	29.4	0.93	23.0	23.8	5.50	5.39	104.4	91	82	453	371
10-60-300-25	2-21-13	105	62	28.7	1.11	23.0	23.1	5.51	5.24	104.6	86	88	325	237
11-60-200-25	2-21-13	123	63	29.0	1.34	23.2	23.4	5.57	5.33	104.9	81	93	232	139
12-60-100-25	2-20-13	137	62	29.2	1.93	22.7	23.7	5.45	5.39	104.8	71	..	....	...
13-40-400-35	2-24-13	93	43	29.4	1.08	18.5	19.6	4.40	4.41	103.7	72	81	472	391
14-40-300-35	2-21-13	102	42	28.9	1.22	17.7	18.4	4.21	4.14	103.9	64	85	328	243
15-40-200-35	2-21-13	107	43	29.0	1.49	18.7	18.8	4.46	4.25	104.2	56	89	246	157
16-40-100-35	2-20-13	130	41	29.2	1.92	17.7	18.3	4.22	4.14	104.2	48	...	.	.
17-100-400-15	3-11-13	138	103	29.3	0.90	30.3	30.3	7.33	6.94	105.6	123	102	443	341
18-100-400-17.5	2-25-13	118	103	29.3	1.00	33.0	33.5	7.99	7.68	105.7	136	98	447	349
19-100-400-20	3-15-13	150	105	28.8	1.30	37.9	37.4	9.13	8.54	105.2	153	114	437	323
20-100-400-20	2-25-13	119	103	29.3	1.26	39.1	38.6	9.42	8.80	105.2	156	98	443	345
21-100-400-25	2-25-13	119	104	29.4	1.65	45.1	44.7	10.82	10.16	104.7	181	98	438	340
22-100-400-25	3-15-13	153	104	28.8	1.74	43.8	45.8	10.45	10.36	104.2	181	112	435	323

Table 6 (Continued)

Observed Values for Engine Tests with Reheated Air

Designation		Venturi Meter				Indicator Data				b. h. p. Data		Reheater		
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
	Date	Temperature of Air Upstream (deg. F.)	Pressure of Air Upstream (lb. per sq. in. Gage)	Barometer (in. Mercury)	Venturi Meter Manometer (in. Mercury)	m. e. p. Head End of Cylinder	m. e. p. Crank End of Cylinder	i. h. p. Head End of Cylinder	i. h. p. Crank End of Cylinder	Revolutions per min.	Weight on Scales	Temperature Entering Reheater (deg. F.)	Temperature Leaving Reheater (deg. F.)	Rise of Temperature (deg. F.)
23-100-400-27.5	2-25-13	120	104	29.4	1.93	48.5	48.7	11.59	11.04	104.4	200	93	445	352
24-100-400-30	3-11-13	124	105	29.3	2.15	49.3	51.4	11.74	11.60	104.0	209	100	439	339
25-100-400-30	3-15-13	151	104	28.8	2.21	50.9	50.9	12.12	11.49	104.0	206	107	435	328
26-100-400-35	3-11-13	137	106	29.3	2.78	57.1	56.1	13.55	12.61	103.6	236	103	439	336
27-100-300-25	3-19-13	135	104	29.1	2.00	43.2	45.5	10.30	10.28	104.1	176	106	331	225
28-100-300-25	4-9-13	139	104	29.1	1.95	42.5	45.8	10.14	10.36	104.2	176	104	323	219
29-100-200-25	3-20-13	138	105	29.0	2.52	44.3	47.2	10.58	10.68	104.3	169	108	213	105
30-100-200-25	4-9-13	124	105	29.1	2.32	42.7	45.9	10.22	10.39	104.3	171	96	209	113
31-100-100-25	4-3-13	135	105	28.9	2.97	41.6	45.3	9.92	10.22	104.1	166	....	....	....
32-80-400-25	3-15-13	141	84	28.8	1.38	33.6	35.7	8.03	8.09	104.4	136	102	448	346
33-80-400-25	4-9-13	126	83	29.1	1.28	32.2	34.8	7.68	7.86	104.1	130	101	437	336
34-80-300-25	3-19-13	109	83	29.1	1.59	33.1	35.6	7.88	8.03	104.0	132	92	334	242
35-80-300-25	4-7-13	114	83	29.4	1.55	32.1	34.2	7.65	7.72	104.1	131	84	321	237
36-80-200-25	3-20-13	125	84	29.0	1.92	32.8	34.8	7.83	7.87	104.2	119	102	215	113
37-80-200-25	4-7-13	103	84	29.4	1.84	32.6	34.4	7.77	7.77	104.1	131	81	218	137
38-80-100-25	3-26-13	139	84	29.2	2.31	31.0	32.6	7.41	7.38	104.4	122	....	....	....
39-80-100-25	4-3-13	129	84	28.9	2.34	32.0	34.8	7.62	7.85	104.0	122	....	....	....
40-60-400-25	3-19-13	95	62	29.3	0.95	23.3	23.9	5.55	5.40	104.1	82	83	458	375
41-60-400-25	3-28-13	106	62	29.5	0.96	22.7	23.9	5.41	5.40	104.1	93	84	452	368
42-60-300-25	3-19-13	101	62	29.3	1.06	22.6	23.6	5.39	5.33	104.1	82	87	327	240
43-60-300-25	3-28-13	103	63	29.5	1.07	22.8	23.8	5.43	5.38	104.1	93	83	347	264
44-60-200-25	3-28-13	115	63	29.5	1.33	22.9	24.0	5.46	5.42	104.0	90	81	206	125
45-60-200-25	4-7-13	96	63	29.4	1.31	22.9	23.7	5.47	5.36	104.3	80	80	220	140
46-60-100-25	3-26-13	132	63	29.2	1.59	22.0	23.9	5.25	5.41	104.3	72	....	....	....
47-60-100-25	4-3-13	122	63	28.9	1.60	22.6	24.2	5.39	5.17	104.1	78		....	....

Table 6 (Continued)

Observed Values for Engine Tests with Reheated Air

		Venturi Meter				Indicator Data				b. h. p. Data		Reheater		
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Designation	Date	Temperature of Air Upstream (deg. F.)	Pressure of Air Upstream (lb. per sq. in. Gage)	Barometer (in. Mercury)	Venturi Meter Manometer (in. Mercury)	m. e. p. Head End of Cylinder	m. e. p. Crank End of Cylinder	i. h. p. Head End of Cylinder	i. h. p. Crank End of Cylinder	Revolutions per min.	Weight on Scales	Temperature Entering Reheater (deg. F.)	Temperature Leaving Reheater (deg. F.)	Rise of Temperature (deg. F.)
48-100-400-25	5-14-13	138	104	29.0	1.55	43.0	45.5	10.26	10.29	104.2	181	116	432	316
49-100-300-25	5-15-13	138	105	29.0	1.81	42.2	45.0	10.07	10.17	104.2	176	116	313	197
50-100-200-25	5-20-13	127	104	29.0	2.18	43.9	44.8	10.45	10.11	104.0	171	107	216	109
51-100-100-25	5-28-13	137	105	29.1	2.80	42.0	45.2	9.98	10.18	103.8	166	....	....	....
52-80-400-25	5-19-13	116	83	29.3	1.14	32.7	34.1	7.80	7.71	104.2	133	96	433	337
53-80-300-25	5-19-13	110	84	29.3	1.36	32.6	34.1	7.77	7.70	104.1	133	97	324	227
54-80-200-25	5-20-13	119	84	29.1	1.67	33.0	33.5	7.86	7.56	104.1	131	99	215	116
55-80-100-25	5-28-13	115	84	29.1	2.18	32.2	35.6	7.67	8.03	104.0	114	.	....	....
56-60-400-25	5-19-13	110	62	29.3	0.81	22.5	23.7	5.37	5.36	104.2	86	93	445	352
57-60-300-25	5-19-13	110	63	29.3	1.03	23.2	24.3	5.53	5.48	104.0	83	94	330	236
58-60-200-25	5-20-13	106	63	29.1	1.25	23.7	24.7	5.64	5.58	104.0	82	93	216	123
59-60-100-25	5-28-13	120	64	29.1	1.58	23.2	25.3	5.53	5.71	104.0	72	...	....	....
60-100-400-15	5-21-13	126	103	29.0	0.76	29.3	30.6	7.07	6.99	105.3	120	114	439	325
61-100-400-20	5-21-13	135	104	29.0	1.11	36.9	39.1	8.85	8.88	104.7	152	118	427	309
62-100-400-30	5-21-13	145	105	29.0	1.97	49.7	51.0	11.82	11.50	103.9	206	122	425	303
63-100-400-35	5-21-13	147	107	29.0	2.56	56.7	56.3	13.41	12.62	103.3	236	124	436	312

23 to 47, inclusive, are the first tests with incomplete expansion; and those numbered 60 to 63, inclusive, are later tests with variable cut-off. Obviously this classification is overlapping. Thus the variable cut-off tests are also incomplete expansion tests. The object in view in tests 17 to 22, inclusive, was the determination of the effects of cut-off when the temperature and pressure were maintained constant at 400 degrees Fahrenheit and 100 pounds per square inch gage, respectively.

16. *Complete Expansion Tests.*—In these tests the load was adjusted to give a cut-off such that the release was at atmospheric pressure and substantially at the end of the stroke; in other words the "toe" of the indicator diagram came to a well defined point and was never rounded or looped. As shown by the designation, these tests were run at four different gage pressures, including 40, 60, 80, and 100 pounds per square inch. Each pressure was combined with four different temperatures, including 100, 200, 300, and 400 degrees Fahrenheit.

It was in the complete expansion tests that freezing of the exhaust first caused trouble. To expand from 100 pounds gage to atmospheric pressure requires a ratio of expansion of about 5.3. The cut-off at this pressure is shown by the indicator diagrams to be at about 15 per cent of the stroke, the release pressure about 1 pound above atmosphere, and the drop of pressure at cut-off about 12 per cent of the initial gage pressure. Then, if subscript 1 denotes conditions at cut-off and subscript 2 those at release, we have the formula:

$$\frac{p_1}{p_2} = \frac{v_2{}^n}{v_1} = \left(\frac{100 + \text{clearance per cent}}{\text{cut-off per cent} + \text{clearance per cent}}\right)^n$$

Taking atmospheric pressure at 14.5 pounds, which was about the average condition for the tests, we have:

$$p_1 = 100 - 12 + 14.5 = 102.5$$
$$p_2 = 14.5 + 1 = 15.5$$

$$\frac{102.5}{15.5} = \left(\frac{105.5}{20.5}\right)^n \text{ or } 6.61 = 5.15^n$$

$$n = \frac{\log.\ 6.61}{\log.\ 5.15} = \frac{0.821}{0.713} = 1.15$$

which checks very well with the value of n obtained by plotting the indicator diagrams taken under these conditions on logarithmic cross-section paper. This value of n shows a considerable heat exchange between the working medium and the cylinder walls. From the formula

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{n-1} = 5.15^{0.15}$$

we find that for air at 100 degrees at cut-off $T_1 = 560$ and $T_2 = 438$, that is, the release temperature was −22 degrees Fahrenheit. It is probable, however, that at cut-off the temperature of the air was at least 20 degrees below the throttle temperature, as the drop in temperature between the throttle and steam chest is 5 degrees for tests of this kind. This would make the temperature at release 37 degrees below zero Fahrenheit. It is not to be inferred that the value of n found above is constant under all conditions. With cut-off at 25 per cent of the stroke, or more, the average value is about $n = 1.275$. In test 4–100–100–15 the temperature registered by the thermometer in the exhaust pipe was −11 degrees Fahrenheit. With a later cut-off and consequently larger amount of air this same thermometer registered 10 to 15 degrees lower than this (see test 31–100–100–25). It might be expected that the lowest temperature would be obtained with earliest cut-off. This would occur if the value of n remained constant under all conditions of operation, but with quantities of air as small as those admitted when the cut-off is at only 15 per cent of the stroke the heat absorbed during the stroke has a greater relative warming effect. In consequence the value of n increases; conversely, the expansion becomes more nearly adiabatic as the cut-off is made later.

Calculating the release temperature when $n = 1.275$, the formula becomes:

$$\frac{T_1}{T_2} = \left(\frac{100 + 5.5}{25 + 5.5}\right)^{0.275}$$

For $T_1 = 560$, this gives $T_2 = 398$, or 62 degrees below zero Fahrenheit. The fall of temperature between throttle and steam chest is not so great in this case as with the 15 per cent cut-off. We may therefore assume that the temperature at cut-off is nearer the throttle temperature but somewhat below it. It is safe to say that the temperature of the air in the cylinder just as the valve is on the point of releasing is between 35 and 50 degrees below zero when the cut-off is at from 15 to 20 per cent of the stroke, and 60 to 70 degrees below zero when the cut-off is at 20 to 25 per cent of the stroke, the temperature at the throttle being in all cases 100 degrees Fahrenheit. After reaching the value 1.275 at 25 per cent cut-off *n* changes very little, so that the release temperature now depends only on the ratio of expansion, and as this diminishes with later cut-off the temperature in the exhaust again rises as the cut-off becomes later than 25 per cent.

With such temperatures prevailing in the cylinder and exhaust pipe, any moisture in the air is certain to cause operating difficulties in a few minutes. The exhaust ports and exhaust pipe become coated with ice, resulting in an increase in the back pressure which may in time prevent operation of the engine. During the tests, vigorous pounding of the exhaust pipe was frequently necessary to clear it of ice. This was done as soon as the indicator diagrams gave evidence, through an increase in the back pressure, of such ice accumulation.

In addition to the trouble from the formation of ice, the low temperature in the cylinder seriously interfered with effective lubrication. The increased viscosity of the lubricant resulting from the low temperature in the cylinder caused a large internal loss by friction. This difficulty was particularly pronounced when a mixture of cylinder oil and graphite was used as the cylinder lubricant. These two effects of the cold caused the brake horse power and mechanical efficiency of the engine to show considerable irregularities.

Various expedients were adopted to eliminate the lubrication difficulties but without complete success in any instance. Figure 11 shows the effect of variable lubrication produced through the infrequent injection of oil into the cylinder. When oil was thus injected it was noticed that a surprisingly large addition of weight was required to keep the brake scales in equilibrium immediately after the injection. This method of feeding oil was therefore immediately abandoned. In Figure 11 the heavy line is the indicator diagram taken just before lubrication; the dotted line is that taken after lubrication. The interval between the two diagrams was less than 10 minutes. No lubricant had been injected for more than one hour prior to taking the first of these diagrams.

At times the combustion in the reheater was extremely imperfect, in which case a third factor affecting the mechanical efficiency was introduced, one whose effect was entirely unmeasurable. When large quantities of unburned carbon with possibly some free gasoline passed into the cylinder, it is probable that the cylinder lubricant became more viscous because of the presence of the carbon, or its lubricating qualities may have been partially destroyed by the presence of free gasoline. To reduce these lubrication difficulties, the engine was finally equipped with a force feed lubricator set to feed 9 drops per minute. With this device cylinder lubrication was improved although it was never entirely satisfactory.

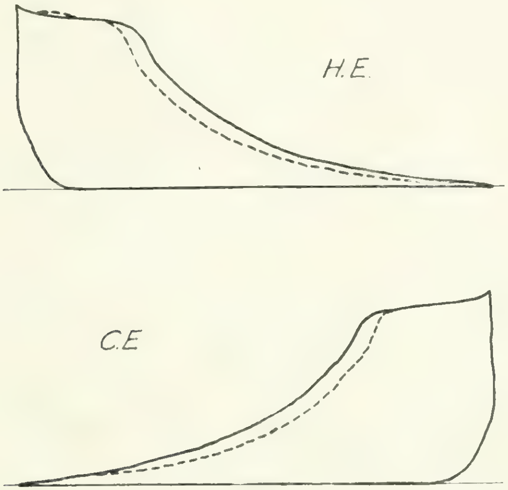

FIG. 11. INDICATOR DIAGRAMS SHOWING EFFECT OF CYLINDER LUBRICATION

The effect of these disturbing conditions can readily be seen by examination of Items 6 and 7 in Table 7, or from the curves of mechanical efficiency, Fig. 12. The maximum and minimum mechanical efficiencies obtained under similar conditions are tabulated together in Tables 9 and 10.

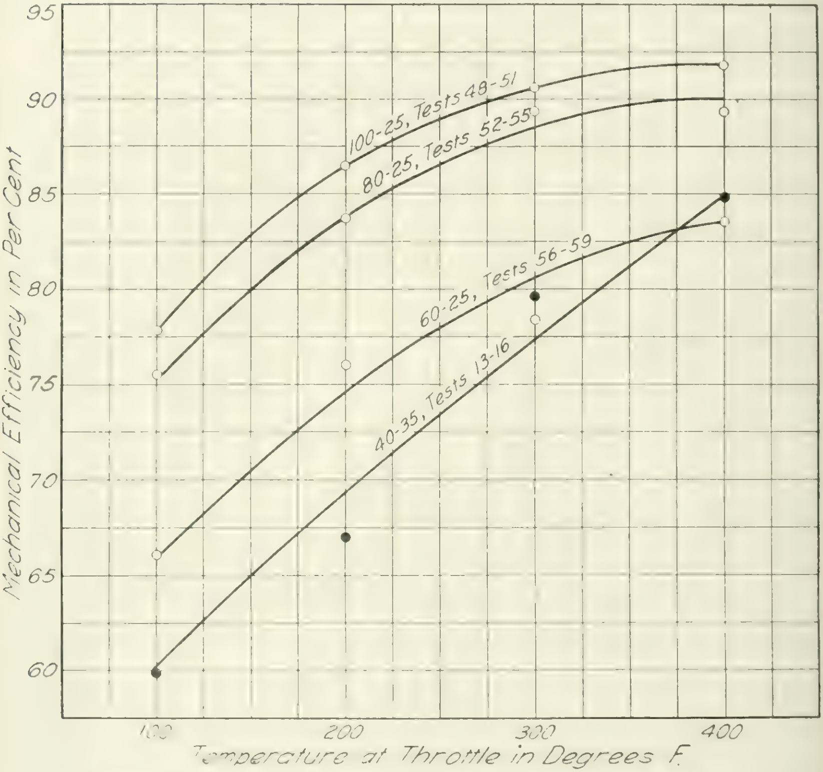

FIG. 12. EFFECT OF CHANGES IN LUBRICATION CONDITIONS

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TABLE 7

OBSERVED VALUES AND COMPUTED RESULTS FOR ENGINE TESTS WITH REHEATED AIR

Designation	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	Designation
			Total Air					Air Rate					Temperature (deg F)																		13,000 B t u Fuel Applied at 60% Efficiency		20,000 B t u Fuel Applied at 80% Efficiency		
	Pressure at Throttle lb per sq in Gage	Temp at Throttle (deg F)	lb per hr	cu ft Free Air per min (60° F, 14 7 lb per sq in Abs)	i h p	b h p	Mech Efficiency	lb per hr per i h p	cu ft Free Air per min per i h p (62° F, 14 7 lb per sq in Abs)	lb per hr per b h p	cu ft Free Air per min per b h p (62° F, 14 7 lb per sq in, Abs)	Average Cut-off Per Cent of Stroke	In Steam Chest	At Exhaust	Room	Barometer (in Mercury)	B t u Req to Raise Air (Item 3) from 100° F to Throttle Temp	lb Air Req per i h p hr at 100° F. and Pressure in Item 1	i h p hr Obtainable from Air (Item 3) at 100° F and Pressure in Item 1	i h p hr Gained (Item 5 Less Item 19)	B t u Req per i h p hr Gained (Item 17 ÷ Item 20)	Thermal Efficiency (2546 ÷ Item 21)	Ideal Efficiency (See Note)	Efficiency Ratio (Item 22 ÷ Item 23)	lb Air Req. per b h p hr at 100° F. and Pressure in Item 1	b h p hr Obtainable from Air (Item 3) at 100° F and Pressure in Item 1	b h p hr Gained (Item 6 Less Item 26)	B t u Req per b h p hr Gained (Item 17 ÷ Item 27)	i h p hr Gained per 1000 B t u Added	b h p hr Gained per 1000 B t u Added	lb per hr per i h p hr Gained	lb per hr per b h p hr Gained	lb per hr per i h p hr Gained	lb per hr per b h p hr Gained	
[illegible]	100	400	505	111	13.66	12.31	90.2	37.1	8.14	41.1	9.00	14.2	336	62	78	29.0	36 400	48.3	10.46	3.20	11 350	22.4	44.4	50.6	60.8	8.31	4.00	9 110	0.088	0.111	1.45	1.16	0.71	0.57	1-100-400-15
[illegible]	100	300	527	114	13.38	11.78	88.1	38.0	8.35	44.8	9.81	14.8	269	54	77	29.0	25 340	48.3	10.91	2.47	10 250	24.9	44.4	56.0	60.8	8.68	3.10	8 170	0.098	0.122	1.31	1.05	0.64	0.52	2-100-300-15
[illegible]	100	200	574	126	13.54	11.25	83.2	42.3	9.27	49.7	10.87	14.5	188	34	76	29.0	13 760	48.3	11.89	1.65	8 370	30.4	44.4	68.5	60.8	9.44	1.81	7 020	0.118	0.142	1.07	0.91	0.53	0.44	3-100-200-15
[illegible]	100	100	684	150	14.11	10.69	75.7	48.5	10.61	64.0	14.03	15.8	95	-11	76	29.2	0	48.3																	4-100-100-15
[illegible]	80	400	481	106	12.66	11.17	88.2	38.0	8.35	43.0	9.44	18.6	297	65	73	29.4	34 600	51.7	9.31	3.35	10 310	24.7	41.3	69.7	68.0	7.08	4.09	8 470	0.097	0.118	1.32	1.08	0.65	0.53	5-80-400-20
[illegible]	80	300	520	115	12.51	10.65	85.1	41.6	9.13	48.8	10.70	18.6	225	39	71	29.4	34 960	51.7	10.03	2.48	10 100	25.2	41.3	61.1	68.0	7.65	3.00	8 320	0.099	0.120	1.29	1.07	0.63	0.52	6-80-300-20
[illegible]	80	200	522	115	12.35	9.10	73.7	42.3	9.27	57.4	12.57	17.5	169	15	64	29.0																			7-80-200-20
[illegible]	80	100	680	149	13.15	9.48	72.0	51.7	11.33	71.7	15.70	21.1	92	-9	73	29.2																			8-80-100-20
[illegible]	60	400	446	99	10.89	9.49	87.1	40.9	8.98	47.0	10.28	24.8	298	83	73	29.4	32 340	54.4	8.22	2.67	12 080	21.1	37.1	56.9	73.1	6.10	3.39	9 440	0.082	0.105	1.55	1.21	0.75	0.60	9-60-400-25
[illegible]	60	300	481	106	10.75	9.00	83.7	44.7	9.80	53.5	11.71	23.8	222	54	70	28.9	23 080	54.4	8.84	1.91	12 070	21.1	37.1	56.9	73.1	6.58	2.42	9 440	0.082	0.105	1.55	1.21	0.75	0.60	10-60-300-25
[illegible]	60	200	522	115	10.90	8.50	78.0	47.9	10.50	61.5	13.46	24.1	169	37	67	29.0	12 520	54.4	9.61	1.29	9 730	26.1	37.1	70.6	73.1	7.05	1.45	8 640	0.103	0.116	1.25	1.11	0.61	0.54	11-60-200-25
[illegible]	60	100	614	135	10.84	7.44	68.6	56.6	12.41	82.5	18.05	24.8	91	7	80	29.2	0	54.4																	12-60-100-25
[illegible]	40	400	416	92	8.81	7.47	84.8	47.2	10.32	55.5	12.20	34.7	301	104	74	29.4	40 280	61.8	6.73	2.08	19 350	13.2	31.3	41.3	104.8	3.97	3.50	11 500	0.051	0.087	2.48	1.47	1.21	0.72	13-40-400-35
[illegible]	40	300	435	96	8.35	6.65	79.6	52.1	11.43	65.4	14.33	32.7	215	64	69	28.9	20 880	61.8	7.03	1.32	15 800	16.1	31.3	51.6	104.8	4.15	2.50	8 360	0.064	0.119	2.02	1.07	0.99	0.53	14-40-300-35
[illegible]	40	200	480	104	8.71	5.83	67.0	55.1	12.08	82.4	18.05	34.0	166	48	68	29.0	11 520	61.8	7.77	0.94	12 250	20.8	31.3	66.4	104.8	4.58	1.25	9 210	0.082	0.109	1.57	1.18	0.76	0.58	15-40-200-35
[illegible]	40	100	524	116	8.36	5.00	59.8	62.7	13.75	104.8	22.97	32.5	95	25	80	29.2	0	61.8																	16-40-100-35
[illegible]	100	400	514	114	14.27	12.98	91.0	36.7	8.02	40.6	8.82	15.7	356	66	69	29.3	37 700	48.2	10.83	3.44	10 920	23.3	44.4	53.6	60.8	8.61	4.37	8 630	0.091	0.116	1.40	1.10	0.68	0.55	17-100-400-15
[illegible]	100	400	561	122	15.67	14.37	91.7	35.9	7.87	39.0	8.55	17.4	340	66	78	29.3	40 380	47.6	11.77	3.90	10 330	24.7	44.4	55.6	58.7	9.56	4.18	9 650	0.097	0.104	1.33	1.24	0.65	[illegible]	[illegible]
[illegible]	100	400	622	136	17.67	16.09	91.1	35.2	7.73	38.6	8.44	20.0	360	64	75	28.8	44 800	47.4	13.17	4.50	9 660	25.5	44.4	57.6	56.9	10.94	5.15	9 700	0.100	0.115	1.28	1.11	0.62	[illegible]	[illegible]
[illegible]	100	400	626	137	18.22	16.42	90.1	34.4	7.54	38.1	8.35	20.3	342	64	76	29.3	45 050	47.4	13.25	4.97	9 080	28.1	44.4	63.3	56.9	11.04	5.38	9 150	0.111	0.110	1.16	1.17	0.57	[illegible]	[illegible]
[illegible]	100	400	715	157	20.98	18.96	90.3	34.1	7.48	37.7	8.26	24.8	344	62	75	29.4	51 500	46.6	15.33	5.65	9 110	28.0	44.4	63.1	54.4	13.15	5.81	8 800	0.110	0.113	1.16	1.13	0.57	[illegible]	21-100-400-25
[illegible]	100	400	715	157	20.81	18.87	90.6	34.3	7.52	37.9	8.30	24.8	363	65	73	28.8	51 500	46.6	15.33	5.48	9 390	27.1	44.4	61.2	54.4	13.15	5.73	8 990	0.107	0.112	1.21	1.15	0.59	0.57	22-100-400-25
[illegible]	100	400	772	169	22.63	20.88	92.2	34.1	7.48	37.0	8.10	27.7	346	56	70	29.4	55 620	46.6	16.57	6.06	9 170	27.8	44.4	62.6	54.4	14.18	6.70	8 300	0.109	0.120	1.17	1.06	0.58	0.52	23-100-400-27
[illegible]	100	400	812	178	23.34	21.73	93.1	34.8	7.63	37.4	8.19	29.7	364	62	71	29.3	58 470	46.6	17.43	5.91	9 900	25.7	44.4	58.0	54.4	14.91	6.82	8 570	0.101	0.116	1.27	1.10	0.62	0.54	24-100-400-[illegible]
[illegible]	100	400	802	176	23.61	21.43	90.7	34.0	7.45	37.5	8.21	29.7	367	64	68	28.8	57 800	46.6	17.23	6.38	9 060	28.2	44.4	63.5	54.4	14.75	6.68	8 650	0.111	0.115	1.15	1.11	0.57	[illegible]	25-100-400-[illegible]
[illegible]	100	400	913	200	26.16	24.47	93.5	34.9	7.63	37.3	8.16	34.4	370	67	69	29.3	65 700	46.6	19.56	6.60	9 960	25.6	44.4	57.9	54.4	16.77	7.70	8 540	0.109	0.117	1.28	1.09	0.62	0.54	26-100-400-[illegible]
[illegible]	100	400	715	157	20.81	18.87	90.6	34.3	7.52	37.9	8.30	24.8	363	65	73	28.8	51 500	46.6	15.33	5.48	9 390	27.1	44.4	61.2	54.4	13.15	5.73	8 990	0.107	0.112	1.21	1.15	0.59	0.57	22-100-400-25
[illegible]	100	300	775	170	20.58	18.32	89.1	37.6	8.24	42.2	9.23	24.6	268	45	77	29.1	37 220	46.6	16.65	3.93	9 725	26.1	44.4	59.0	54.4	14.23	4.09	9 110	0.103	0.109	1.25	1.16	0.61	0.57	27-100-300-25
[illegible]	100	300	763	168	20.50	18.33	89.4	37.3	8.18	41.6	9.10	25.5	266	35	66	29.1	36 650	46.6	16.38	4.12	8 900	28.7	44.4	64.6	54.4	14.03	4.30	8 530	0.113	0.117	1.13	1.09	0.56	0.54	28-100-300-25
[illegible]	100	200	866	189	21.26	17.63	82.9	40.7	8.92	49.0	10.73	25.0	181	14	66	29.0	20 780	46.6	18.58	2.68	7 755	32.8	44.4	74.1	54.4	15.90	1.73	12 000	0.129	0.084	[illegible]	1.55	0.48	0.75	29-100-200-25
[illegible]	100	200	843	184	20.61	17.84	86.5	40.9	8.96	47.2	10.32	25.2	184	0	66	29.1	20 240	46.6	18.10	2.51	8 075	31.6	44.4	71.2	54.4	15.49	2.35	8 630	0.124	0.116	1.03	1.10	0.50	0.54	30-100-200-25
[illegible]	100	100	937	205	20.14	17.29	85.4	46.6	10.20	54.2	11.86	25.5	100	-26	82	28.9	0	46.6																	31-100-100-25
32-80-400-25	80	400	586	129	16.12	14.19	88.0	36.4	7.97	41.3	9.04	25.4	357	73	74	28.8	42 200	50.5	11.60	4.52	9 330	27.3	41.3	66.0	60.8	9.65	4.54	9 300	0.107	0.108	1.20	1.20	0.59	0.58	32-80-400-25
33-80-400-25	80	400	573	126	15.54	13.53	87.2	36.8	8.06	42.3	9.27	25.3	342	50	73	29.1	41 230	50.5	11.33	4.21	9 800	26.0	41.3	63.1	60.8	9.43	4.10	10 060	0.102	0.099	1.26	1.29	0.61	0.63	33-80-400-25
34-80-300-25	80	300	645	141	15.91	13.73	86.3	40.6	8.90	46.9	10.29	25.1	263	51	78	29.1	30 960	50.5	12.77	3.14	9 860	25.8	41.3	62.6	60.8	10.60	3.13	9 900	0.101	0.101	1.27	1.27	0.62	0.62	34-80-300-25
35-80-300-25	80	300	643	144	15.37	13.64	88.7	41.2	9.05	46.4	10.18	24.6	254	36	70	29.4	30 390	50.5	12.52	2.85	10 660	23.9	41.3	57.9	60.8	10.42	3.22	9 440	0.094	0.106	1.37	1.22	0.67	0.59	35-80-300-25
36-80-200-25	80	200	699	154	15.70	12.40	79.0	44.5	9.76	56.4	12.36	24.9	180	22	70	29.0	16 790	50.5	13.84	1.86	9 025	28.2	41.3	68.3	60.8	11.51	0.89	18 850	0.111	0.053	1.14	2.42	0.57	1.17	36-80-200-25
37-80-200-25	80	200	696	152	15.54	13.64	87.8	44.8	9.83	51.0	11.16	25.1	185	17	68	29.4	16 680	50.5	13.79	1.75	9 550	26.6	41.3	64.5	60.8	11.45	2.19	7 620	0.105	0.131	1.23	0.98	0.60	0.47	37-80-200-25
38-80-100-25	80	100	757	166	14.79	12.74	86.2	51.2	11.21	59.3	12.99	24.2	89	-38	70	29.2	0	50.5																	38-80-100-25
39-80-100-25	80	100	765	168	15.47	12.69	82.0	49.4	10.82	60.4	13.21	25.1	95	-12	82	28.9	0	50.5																	39-80-100-25
40-60-400-25	60	400	451	99	10.95	8.54	78.0	41.2	9.05	52.8	11.56	24.9	321	81	73	29.3	32 500	54.4	8.31	2.64	12 300	20.7	37.1	56.0	73.2	6.17	2.37	13 700	0.082	0.073	1.58	1.75	0.77	0.86	40-60-400-25
41-60-400-25	60	400	450	99	10.81	9.68	89.5	41.6	9.13	46.4	10.18	25.2	338	74	71	29.5	32 450	54.4	8.28	2.53	12 810	19.8	37.1	53.5	73.2	6.16	3.52	9 220	0.078	0.109	1.64	1.18	0.80	0.58	41-60-400-25
42-60-300-25	60	300	473	104	10.72	8.54	79.6	44.1	9.69	55.4	12.13	24.8	260	66	77	29.3	22 700	54.4	8.69	2.03	11 180	22.7	37.1	61.2	73.2	6.46	2.08	10 920	0.089	0.091	1.43	1.40	0.70	0.68	42-60-300-25
43-60-300-25	60	300	477	104	10.81	9.68	89.5	44.0	9.67	49.2	10.77	25.1	269	64	68	29.5	22 900	54.4	8.79	2.02	11 340	22.4	37.1	60.4	73.2	6.52	3.16	7 250	0.088	0.139	1.45	0.93	0.71	0.45	43-60-300-25
44-60-200-25	60	200	536	115	10.88	9.36	86.0	48.2	10.56	56.1	12.29	24.9	185	37	62	29.5	12 610	54.4	9.66	1.22	10 320	24.7	37.1	66.6	73.2	7.18	2.18	5 780	0.097	0.173	1.32	0.74	0.65	0.36	44-60-200-25
45-60-200-25	60	200	540	116	10.83	8.34	76.9	48.9	10.72	63.4	13.90	24.1	182	32	69	29.4	12 700	54.4	9.73	1.10	11 550	22.0	37.1	59.5	73.2	7.24	1.10	11 530	0.086	0.086	1.48	1.48	0.72	0.72	45-60-200-25
46-60-100-25	60	100	566	124	10.66	7.51	70.5	53.1	11.61	75.4	16.51	25.3	88	-10	59	29.2	0	54.4																	46-60-100-25
47-60-100-25	60	100	573	126	10.86	8.12	74.7	52.8	11.57	70.5	15.45	24.2	97	10	80	28.9	0	54.4																	47-60-100-25
48-100-400-25	100	400	683	149	20.55	18.87	91.8	33.3	7.30	36.2	7.91	23.9	344	65	93	29.0	49 200	45.3	15.04	5.51	8 920	28.6	44.4	64.4	54.4	12.55	6.32	7 790	0.112	0.128	1.14	1.00	0.56	0.49	48-100-400-25
49-100-300-25	100	300	738	162	20.24	18.33	90.6	36.5	8.00	40.2	8.80	24.5	268	51	91	29.0	35 440	45.3	16.30	3.94	8 990	28.3	44.4	63.9	54.4	13.55	4.78	7 420	0.111	0.135	1.15	0.95	0.57	0.46	49-100-300-25
50-100-200-25	100	200	816	178	20.56	17.79	86.5	39.7	8.69	45.8	10.05	24.6	187	23	89	29.0	19 580	45.3	18.00	2.56	7 640	33.3	44.4	75.2	54.4	15.00	2.79	7 010	0.131	0.143	0.98	0.90	0.48	0.44	50-100-200-25
51-100-100-25	100	100	914	200	20.16	15.69	77.8	45.3	9.94	58.3	12.78	24.4	98	-24	85	29.1	0	45.3	20.16																51-100-100-25
52-80-400-25	80	400	546	119	15.51	13.86	89.3	35.2	7.73	39.5	8.64	23.8	337	66	81	29.3	39 300	48.0	11.37	4.14	9 490	26.8	41.3	64.9	60.8	8.98	4.88	8 060	0.105	0.123	1.22	1.03	0.60	0.50	52-80-400-25
53-80-300-25	80	300	598	131	15.47	13.85	89.4	38.6	8.47	43.2	9.46	23.8	267	54	82	29.3	28 690	48.0	12.43	3.02	9 510	26.8	41.3	64.9	60.8	9.83	4.02	7 130	0.105	0.141	1.22	0.92	0.60	0.44	53-80-300-25
54-80-200-25	80	200	657	144	15.42	12.90	83.7	42.5	9.33	50.8	11.11	24.0	184	27	86	29.1	15 710	48.0	13.67	1.75	8 980	28.3	41.3	68.6	60.8	10.80	2.10	7 480	0.111	0.134	1.15	0.96	0.56	0.46	54-80-200-25
55-80-100-25	80	100	752	165	15.70	11.86	75.5	47.9	10.48	63.3	13.88	24.7		-4	90	29.1	0	48.0	15.70																55-80-100-25
56-60-400-25	60	400	411	90	10.73	8.96	83.5	38.3	8.40	45.8	10.05	24.1	321	83	85	29.3	29 610	51.1	8.05	2.68	11 030	23.1	37.1	62.4	73.1	5.62	3.34	8 880	0.090	0.113	1.41	1.14	0.69	0.56	56-60-400-25
57-60-300-25	60	300	464	102	11.01	8.63	78.4	42.0	9.22	53.8	11.78	25.0	259	64	84	29.3	22 250	51.1	9.08	1.93	11 500	22.1	37.1	59.8	73.1	6.34	2.29	9 730	0.087	0.103	1.47	1.25	0.72	0.61	57-60-300-25
58-60-200-25	60	200	514	112	11.22	8.52	76.0	45.7	10.10	60.3	13.21	24.4	180	39	84	29.1	12 300	51.1	10.05	1.17	10 500	24.3	37.1	65.5	73.1	7.02	1.50	8 200	0.095	0.122	1.35	1.05	0.66	0.52	58-60-200-25
59-60-100-25	60	100	573	126	11.24	7.49	66.1	51.0	11.16	76.4	16.73	24.4		16	90	29.1	0	51.1	11.24																59-60-100-25
60-100-400-15	100	400	487	107	14.06	12.64	89.2	34.6	7.59	38.5	8.39	14.1	340	71	87	29.0	35 120	43.8	10.45	3.61	9 740	26.1	44.4	58.8	60.8	8.02	4.62	7 600	0.103	0.132	1.25	0.97	0.61	0.47	60-100-400-15
61-100-400-20	100	400	586	128	17.73	15.92	89.8	32.9	7.21	36.7	8.03	19.2	347	70	87	29.0	42 040	45.8	12.75	4.98	8 460	30.1	44.4	67.9	56.9	10.28	5.64	7 480	0.122	0.134	1.08	0.96	0.53	0.46	61-100-400-20
48-100-400-25	100	400	683	149	20.55	18.87	91.8	33.3	7.30	36.2	7.91	23.9	344	65	93	29.0	49 200	45.3	15.04	5.51	8 920	28.6	44.4	64.4	54.4	12.55	6.32	7 790	0.112	0.128	1.14	1.00	0.56	0.49	48-100-400-25
62-100-400-30	100	400	765	168	23.32	21.40	91.8	32.9	7.21	35.9	7.86	28.7	355	71	87	29.0	55 200	45.3	16.90	6.42	8 590	29.6	44.4	66.7	54.4	14.10	7.30	7 580	0.116	0.132	1.10	0.97	0.54	0.47	62-100-400-30
63-100-400-35	100	400	874	192	26.03	24.40	93.7	33.6	7.36	35.9	7.86	33.5	357	71	87	29.0	62 850	45.3	19.25	6.78	9 260	27.4	44.4	61.8	54.4	16.06	8.34	7 540	0.108	0.133	1.18	0.97	0.58	0.47	63-100-400-35

NOTE:

$$\text{Item 23, Ideal Efficiency} = 1 - \left[\frac{14.7}{(\text{Item 1}) + 14.7}\right]^{0.286}$$

TABLE 8

COMPARISON OF RESULTS OBTAINED IN TESTS IN MARCH AND MAY RESPECTIVELY

	1	2	3	4	5	6	7	8	9	10
Designation	Date	Cut-off, % of Stroke	Total Air (lb. per hr.)	i. h. p.	b. h. p.	Mech Efficiency	Air Rate (lb. per i. h. p. hr.)	Air Rate (lb. per b. h. p. hr.)	Exhaust Temp. (deg. F.)	Room Temp. (deg. F.)
17-100-400-15	3-11-13	15.7	514	14.27	12.98	91.0	36.7	40.6	66	69
60-100-400-15	5-21-13	14.1	487	14.06	12.64	89.2	34.6	38.5	71	87
19-100-400-20	3-15-13	20.0	622	17.67	16.09	91.1	35.2	38.6	64	78
64-100-400-20	5-21-13	19.2	586	17.73	15.92	89.8	32.9	36.7	70	87
22-100-400-25	3-15-13	24.8	715	20.81	18.87	90.6	34.3	37.9	65	73
48-100-400-25	5-14-13	23.9	683	20.55	18.87	91.8	33.3	36.2	65	93
25-100-400-30	3-15-13	29.7	802	23.61	21.43	90.7	34.0	37.5	64	68
62-100-400-30	5-21-13	28.7	765	23.32	21.40	91.8	32.9	35.9	71	87
26-100-400-35	3-11-13	34.4	913	26.16	24.47	93.5	34.9	37.3	67	69
6[illegible]-100-400-35	5-21-13	33.5	874	26.03	24.40	93.7	33.6	35.9	71	87

TABLE 9

EXTREME VARIATIONS OF MECHANICAL EFFICIENCY WITH SIMILAR PRESSURE AND TEMPERATURE CONDITIONS

Cut-off at 25 per cent of stroke

400 deg.		300 deg.		200 deg.		100 deg.	
High	Low	High	Low	High	Low	High	Low
100 lb. Gage							
91.8	90.3	90.6	89.1	86.5	82.9	85.4	77.8
80 lb. Gage							
89.3	87.2	89.4	86.3	87.8	79.0	86.2	75.5
60 lb. Gage							
89.5	78.0	89.5	78.4	86.0	76.0	74.7	66.1

TABLE 10

EXTREME VARIATIONS OF MECHANICAL EFFICIENCY WITH SIMILAR PRESSURE AND TEMPERATURE CONDITIONS

Various Cut-offs*

400 deg.		300 deg.		200 deg.		100 deg.	
High	Low	High	Low	High	Low	High	Low
100 lb. Gage							
93.7 33.5	90.2 14.2	90.6 24.5	88.1 14.8	86.5 25.2	82.9 25.0	85.4 25.5	75.7 15.8
80 lb. Gage							
89.3 23.8	87.2 25.3	89.4 23.8	85.1 18.6	87.8 25.1	73.7 17.5	86.2 24.2	72.0 21.1

TABLE 10 (CONTINUED)

EXTREME VARIATIONS OF MECHANICAL EFFICIENCY WITH SIMILAR PRESSURE AND TEMPERATURE CONDITIONS

Various Cut-offs*

60 lb. Gage

89.5 25.2	78.0 24.9	89.5 25.1	78.4 25.0	86.0 24.9	76.0 24.4	74.7 24.2	66.1 24.4

40 lb. Gage

84.8 34.7	79.6 32.7	67.0 34.0	59.8 32.5

* The lower figures under each pressure are the actual points of cut-off in per cent of the stroke.

17. *Variable Cut-off Tests.*—Tests 17 to 22 and 60 to 63 inclusive were made for the purpose of determining the air economy of the engine with different ratios of expansion, other conditions remaining constant. All these tests were therefore made with the same pressure and temperature conditions at the throttle, namely 100 pounds per square inch gage pressure and 400 degrees Fahrenheit. The cut-off was increased from test to test by about 5 per cent, starting with 15 per cent in the first test of each series. At the assigned initial pressure this is the earliest cut-off that could be maintained, since this gives atmospheric pressure at release and any earlier cut-off would have caused a loop of negative work on the diagram.

18. *Incomplete Expansion Tests.*—Of the 3 tests made on compressed air 34 were incomplete expansion tests, the cut-off in each case being at 25 per cent of the stroke. Since the cut-off was kept constant during these tests the lowest initial pressure which could be maintained without the formation of a loop of negative work on the indicator diagram was 60 pounds per square inch gage.

As shown by Table 7, these tests include three initial pressures, 100, 80, and 60 pounds per square inch gage, each combined with four temperatures at the throttle, 400, 300, 200, and 100 degrees Fahrenheit, respectively. With two exceptions the tests were run under identical conditions; the exceptions being those tests made with a pressure of 100 pounds gage and air temperatures of 400 degrees and 100 degrees Fahrenheit. Tests 27 to 47 inclusive were made during March and April, while tests 48 to 63 inclusive were made during the latter part of May. This repetition of tests under the same conditions seemed desirable when it was observed that there was a perceptible decrease in the air consumption per indicated horse power as the season advanced. The curves of air rate per indicated horse power per hour plotted against temperature for tests made during one month fell slightly below those made during the preceding month in spite of every effort to make all conditions the same. The May tests gave an air rate per indicated horse power per hour of from 1 to 1½ pounds less than those made in March. Table 8 shows these variations. With a cut-off earlier than 25 per cent the variations in air consumption were even greater than those with 25 per cent cut-off.

Since the engine employed in these tests was new, the gradual reduction in air consumption was largely brought about by improvement in the condition of the cylinder and valves, resulting from wear; and to a lesser extent by the increasing room temperature as the outdoor temperature increased. Thus, during the tests made in May the room temperature was about 20 degrees higher than that maintained in Macrh.

Table 7 presents certain observed and all computed values for the whole series of air-engine tests. While the derivation of the various items is generally obvious, it seems desirable to indicate the source of the data or the method of computation of each item listed.

Items 1 and 2 are average values of data from tests.

Item 3 is computed from the Venturi meter data given in Table 6, using the general method described in the paper mentioned on page 20.

Item 4=Item 3 × 0.219 (see Section 8).

Item 5 is computed from the engine constants, L = stroke in feet and A = mean area of piston in square inches, the average speed in revolutions per minute, N, and the average mean effective pressure, P, from the various indicator diagrams, by the formula

$$\text{I. h. p.} = \frac{2\ PLAN}{33000}$$

Item 6 is computed from the radius of brake arm in feet, R, the load on the scales in pounds, W, and the average revolutions per minute, N, by the formula

$$\text{B. h. p} = \frac{2\,\pi\,R\,W\,N}{33000}\,.$$

Item 7 = Item 6 divided by Item 5.
Item 8 = Item 3 divided by Item 5.
Item 9 = Item 4 divided by Item 5.
Item 10 = Item 3 divided by Item 6.
Item 11 = Item 4 divided by Item 6.

Item 12 is obtained from the indicator diagrams.

Items 13, 14, 15, and 16 are average data from the tests.

Item 17 = Item 3 multiplied by the difference between Items 2 and 1 and by the specific heat of air at constant pressure (0.24), the values recorded being the nearest tens.

Item 18 is selected from the curves representing the air consumption per indicated horse power for various throttle temperatures, Figs. 13 and 14, since these values are possibly somewhat more consistent than the results of the individual test at 100 degrees temperature at the throttle; wherefore these values do not agree exactly with the items given under Item 8.

Item 19 = Item 3 divided by Item 18.
Item 20 = Item 5 minus Item 19.
Item 21 = Item 17 divided by Item 20.

Item 22 = the heat equivalent of one horse power per hour (2546 B.t.u.) divided by Item 21.

$$\text{Item 23} = 1 - \left(\frac{14.7}{\text{Item 1} + 14.7}\right)^{0.286}$$

No attempt has been made to determine the ideal efficiency of the tests based upon incomplete expansion of the air. For some of the tests the value of Item 23 is therefore somewhat high.*

Item 24 = Item 22 divided by Item 23. For the reason stated under Item 23, the value of Item 24 is in some cases slightly smaller than would have been the case had the ideal efficiency for incomplete expansion been computed.

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[*] See Section 5, Case II, for complete adiabatic expansion, which represents the maximum attainable thermal efficiency.

Item 25 is obtained from the curves representing the air consumption per brake horse power per hour for the various throttle temperatures. These values were selected for the reason stated under Item 18.

Item 26 = Item 3 divided by Item 25.

Item 27 = Item 6 minus Item 26.

Item 28 = Item 17 divided by Item 27.

Item 29 = Item 20 multiplied by 1000 and divided by Item 17.

Item 30 = Item 27 multiplied by 1000 and divided by Item 17.

Item 31 = 1000 divided by the product of 13 000 and 0.6 and Item 29.

Item 32 = 1000 divided by the product of 13 000 and 0.6 and Item 30.

Item 33 = 1000 divided by the product of 20 000 and 0.8 and Item 29.

Item 34 = 1000 divided by the product of 20 000 and 0.8 and Item 30.

The steam chest temperatures, Item 13, show a surprising drop between the temperature at the throttle and the point in the steam-chest where these temperatures were taken. The amount of drop decreases rapidly with the lowering of the throttle temperature. As the steam chest thermometer was placed in a thermometer well, it is probable that in the first twenty-one tests these differences are too great. This well was later removed and replaced by a stuffing-box through which the thermometer was passed, thus giving the temperature by direct contact between the bulb and the heated air as it passed to the steam ports of the engine. The drop of temperature is greater the earlier the cut-off, and decreases with later cut-off. For cut-off at 25 per cent of the stroke a difference of 40 to 50 degrees may be expected between throttle and steam chest temperatures when the throttle conditions are 100 pounds gage pressure and 400 degrees Fahrenheit. Exhaust temperatures, Item 14, are particularly unsatisfactory. The thermometer could not be placed near enough to the exhaust port of the engine to give the reading there and the temperature in the exhaust pipe was affected both by the room temperature and the exhaust pipes of other engines.

The heat required to raise the weight of air used in a particular test from 100 degrees Fahrenheit to the temperature maintained at the throttle, Item 17, does not include the loss of heat from the air pipe between the reheater and the throttle. Due to its low specific heat and heat content, the drop in temperature of air flowing in a pipe is large. If the reheater is located l feet from the throttle valve, the temperature, t_r, required at the reheater to maintain a temperature, t_t, at the throttle may be computed by the following equation:

$$log_e\ (t_r - t_o) = \frac{\pi D\ k\ l}{1\ 2\ M\ c_p} + log_e\ (t_t - t_o)$$

in which

t_0 = temperature of the freezing point, 32 degrees Fahrenheit;
d = diameter of pipe in inches;
l = length of pipe in feet;
m = weight of air flowing per hour;
c_p = specific heat of air at constant pressure, (0.24);
k = coefficient of heat loss in B.t.u.per square foot of surface per degree difference between the air inside the pipe and that outside; approximately 3 for uncovered pipe and 0.6 for covered pipe.

The solution of a specific problem by means of the foregoing equation will show the great importance of locating the reheater as near the engine as possible, and of carefully insulating the connecting pipe and fittings.

The air-rates per indicated horse power per hour for various throttle temperatures are shown in Figs. 13 and 14. The former includes the data from tests 1 to 16 inclusive and 22 to 47 inclusive; the latter, tests 48 to 59 inclusive, made after the engine was well "worn in" and with a room temperature 20 degrees higher than in the earlier tests. The air rate in pounds per indicated horse power per hour for various throttle temperatures and gage pressures is shown in Fig. 15. The air rates per brake horse power per hour for various throttle temperatures are shown in Fig. 16. Some of these data are shown in an even more interesting manner in Fig. 12 in which the various pressure curves are drawn with mechanical efficiencies as ordinates and throttle temperatures as abscissæ.

It was expected that there would be some improvement in the mechanical efficiency with increasing air temperatures, but the extent of this improvement was such as to astonish everyone concerned with the tests. With a throttle temperature of 400 degrees Fahrenheit and

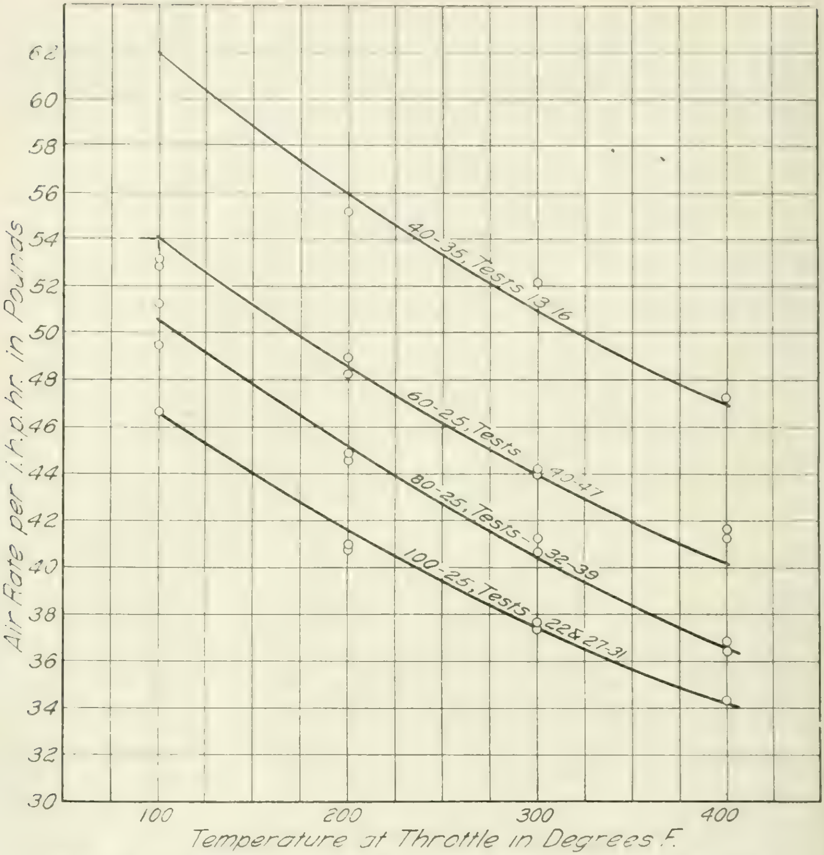

FIG. 13. AIR CONSUMPTION PER INDICATED HORSE POWER PER HOUR FOR VARIOUS THROTTLE TEMPERATURES

a pressure of 100 pounds gage, mechanical efficiencies of 89.2 to 93.7 per cent were obtained; and with a throttle temperature of 100 degrees Fahrenheit and the same pressure, the mechanical efficiencies varied from 75.7 to 85.4 per cent. The lowest recorded mechanical efficiency is that of test 16, its value being only 59.8 per cent. It should be noted that in nearly every test the gain in brake horse power through reheating is greater than the gain in indicated horse power. Tables 9 and 10 present the high and low mechanical efficiencies obtained under various conditions of operation.

Items 31 and 32 (Table 7) present the computed weights of fuel consumed per hour for each indicated and brake horse power gained by reheating. Because of conditions discussed in the preceding paragraph, the weight of fuel per brake horse power per hour is *less* than that required per indicated horse power. While the weight of fuel required per brake horse power per hour fluctuated considerably for different tests, it should be noted that in several tests values of less than one pound of fuel were obtained. Few devices for converting heat into work can show equally satisfactory results.

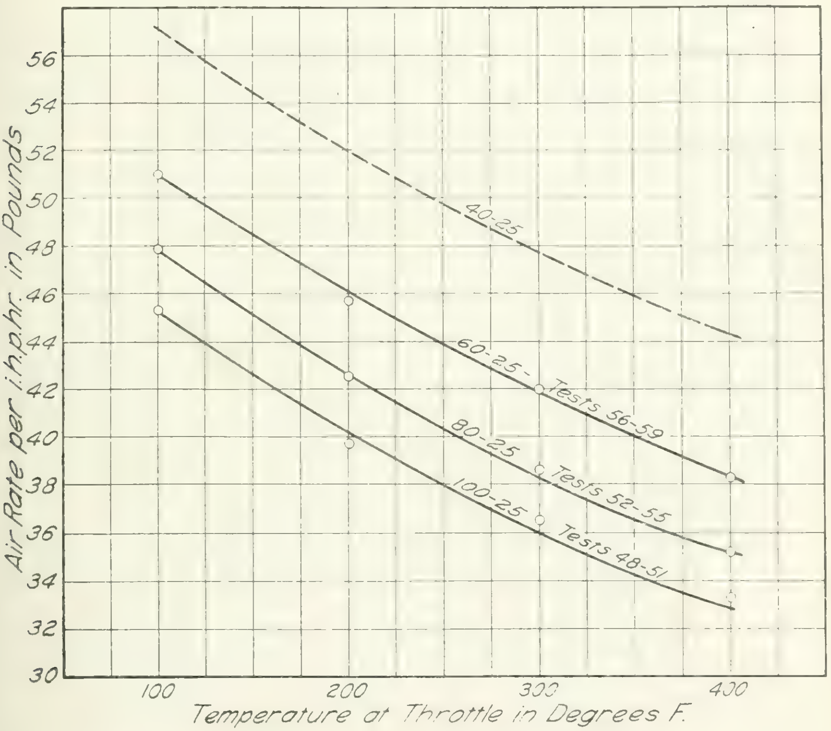

FIG. 14. AIR CONSUMPTION PER INDICATED HORSE POWER PER HOUR FOR VARIOUS THROTTLE TEMPERATURES

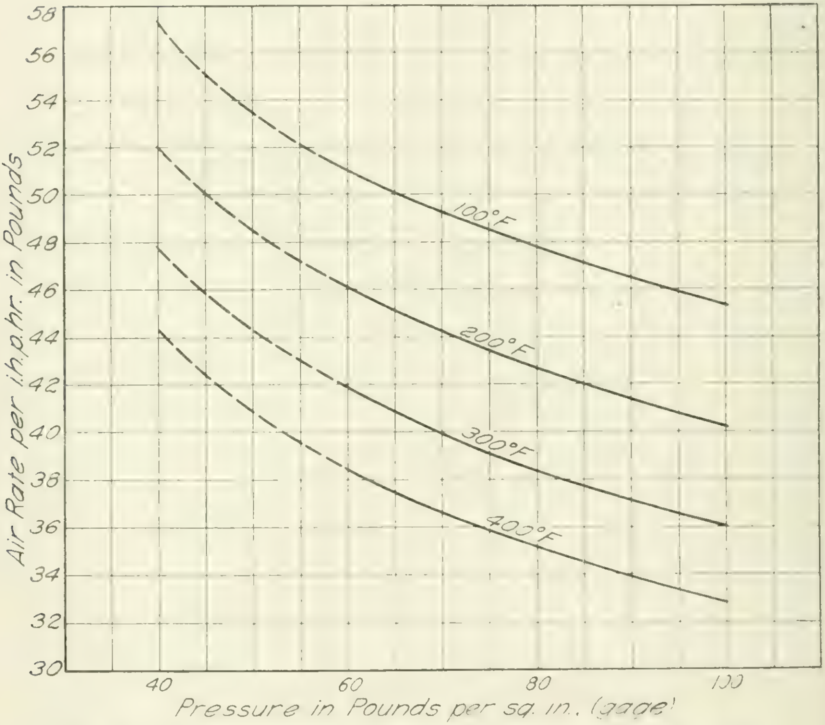

Fig. 15. Air Consumption per Indicated Horse Power per Hour for Various Throttle Temperatures and Gage Pressures

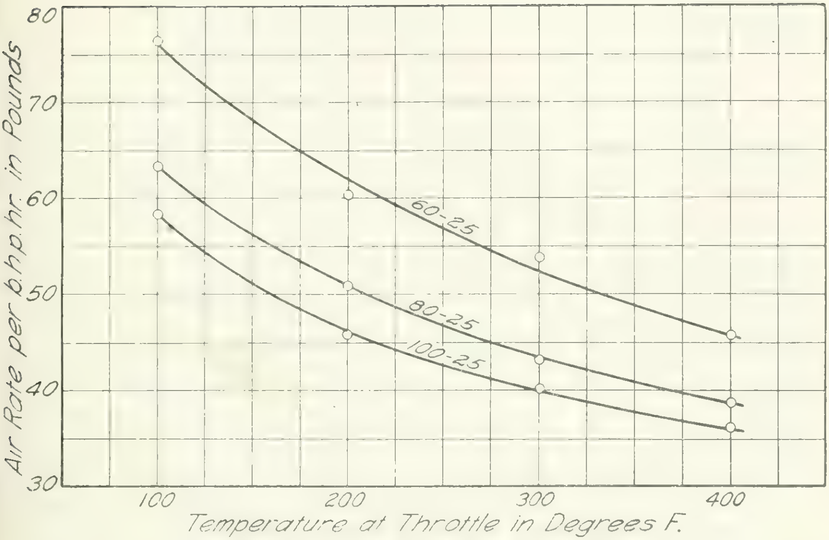

FIG. 16. AIR CONSUMPTION PER BRAKE HORSE POWER PER HOUR FOR VARIOUS THROTTLE TEMPERATURES

V. TESTS OF ENGINE USING MIXTURES OF AIR AND STEAM AS THE WORKING FLUID

19. *Purposes of Tests.*—The primary purpose of this series of tests of air-steam mixtures was the determination of the efficiency resulting from the use of steam as a medium for reheating compressed air; the secondary purpose was the devising of methods of testing which might be applicable in a more complete study of the subject, to be undertaken after the present general investigation.

20. *Method of Conducting Tests.*—While the plant described in Section IV was employed in these tests, with the condenser in service, the use of steam necessitated several changes in the methods of testing from those employed with air. The air from the compressor was delivered to the mixing chamber immediately above the throttle valve at a temperature which was maintained as nearly as possible at 100 degrees Fahrenheit; at this point steam was injected into the air until the required conditions of pressure and ratio of air to steam were obtained. To secure these conditions the running of from one to three preliminary tests was necessary. Before making these trial tests, an estimate was made to determine the approximate weight of air and steam required for a given mixture with the engine cut-off at 25 per cent of the stroke. The valve in the air line was regulated until the Venturi meter showed the correct flow of air, when steam was admitted in sufficient quantity to give the required pressure at the throttle. The plant was operated with these valve adjustments until the condenser was discharging steadily and other conditions were constant. When the weight of condensate showed the desired ratio of air and steam the test was started and continued for one hour.

In this test series much less time was required for securing thermal equilibrium in the cylinder and steam chest than in the air tests. The thermometers at the throttle, in the steam chest, and in the exhaust pipe quickly reached definite constant temperatures which were maintained as long as conditions were kept uniform. Thus, despite the fact that considerable time was required for the adjustment of the mixture, the preparatory run was generally no longer than when air alone was used.

The pressure at the throttle and the load on the brake were kept constant by frequent adjustment throughout the test. The following observations were taken as indicated:

(1) Temperature at throttle (every 10 minutes)
(2) Temperature in steam chest (every 10 minutes)
(3) Revolutions per minute (every 10 minutes)
(4) Pressure in mixing chamber (every 10 minutes)
(5) Temperature in mixing chamber (every 10 minutes)
(6) Temperature of air before entering mixing chamber (every 10 minutes)
(7) Temperatures in throttling steam calorimeter (every 10 minutes)
(8) Venturi manometer (every 2 minutes)
(9) Pressure of air before entering mixing chamber (every 10 minutes)
(10) Temperature in exhaust pipe (every 10 minutes)
(11) Temperature of condensate (every 10 minutes)
(12) Room temperature (taken twice per test)
(13) Barometric pressure (taken once per test)
(14) Weight of condensate from condenser
(15) Weight of condensate from mixing chamber

The total steam admitted is determined in three parts: first, that which is condensed in the mixing chamber and there separated from the remaining mixture; second, that which is condensed in the condenser; and third, that which is contained in the air saturated with water vapor leaving the condenser at atmospheric pressure and the temperature maintained in the condenser. With an ample supply of cold condensing water and a consequent low exhaust temperature, the actual weight of moisture in the exhaust from the condenser is small. For great precision in measuring the steam supplied, the weight of vapor in the air entering the mixing chamber should be subtracted from the total weight of steam determined as described above. Because of difficulty in determining the exact condition of the compressed air in the air main, and the fact that the error in weight of steam should be small even though the air was initially saturated, the air for the sake of simplicity was assumed to be dry. With the air supply at 75 pounds per square inch absolute and 100 degrees Fahrenheit, the error due to this assumption is approximately

2 per cent of the weight of vapor in the exhaust air at 200 degrees Fahrenheit and a small fraction of one per cent of the total weight of steam used in the process.

21. *Discussion of Observed and Calculated Data.*--The heat content per pound of steam added was determined from the calorimeter data Items 7 and 8, Table 11, and recorded as Item 10; this multiplied by the total condensate, Item 15, of the same table, which is the sum of the condensate from the condenser, the condensate from the separator, and the moisture carried away in the air, gives the total heat energy in the steam added. While it may seem that Item 4 of Table 12, which gives the total steam in terms of equivalent dry steam at the throttle pressure, should be obtained by dividing the total heat energy in the steam by the heat content of dry steam at throttle pressure, this is not the case. Although in all of these computations the air was supposed to be delivered to the mixer at 100 degrees Fahrenheit, Item 6, Table 11, shows that this was not actually true. A correction was therefore necessary. If the air entered the mixer below 100 degrees Fahrenheit more steam was actually used than if it had entered at the standard temperature; hence the heat required to raise the air used in the test from the temperature entering the mixer to 100 degrees Fahrenheit was subtracted from the total heat of the steam used. Similarly when the air entered the mixer at a temperature above 100 degrees Fahrenheit less steam was actually used than when the air temperature was 100 degrees; hence the correction was added. The total heat thus corrected and divided by the heat content of dry steam at throttle pressure gives Item 4 of Table 12. By means of this correction it is possible in subsequent calculations to disregard the actual temperature of the air entering the mixer. From the previous air tests, air-rates per indicated horse power and brake horse power are known.

The manometer readings, Item 4, Table 11, are very small when the per cent of steam in the mixture is large. When the air consumption was small the mercury in the manometer was replaced by water which clearly showed small differences of pressure. In the table these pressures were converted from the observed heights of the column of water to heights of a column of mercury. From Item 6 of Table 12 it is seen that after all corrections were made the ratio of the steam used to the total mixture often deviated considerably

TABLE 11

OBSERVED VALUES FOR ENGINE TESTS WITH AIR-STEAM MIXTURES

		Venturi Meter					Throttling Calorimeter			
	1	2	3	4	5	6	7	8	9	10
Designation	Date	Temperature of Air Upstream (deg. F.)	Pressure of Air Upstream (lb. per sq. in. Gage)	Venturi Meter Manometer (in. Mercury)	Barometer (in. Mercury)	Temperature of Air Entering Mixer (deg. F.)	Pressure at Calorimeter (lb. per sq. in Gage)	Temperature at Calorimeter (deg. F.)	Quality of Steam	Heat Content of Steam B. t. u
A-S-100-10	5-5-13	128.6	101.3	1.42	29.31	106.7	121.4	250.4	0.972	1168
A-S-100-20	5-5-13	137.4	101.0	0.91	29.30	113.4	122.1	251.0	0.972	1168
A-S-100-30	5-5-13	136.0	102.9	0.64	29.30	113.0	120.1	249.9	0.971	1167
A-S-100-50	5-6-13	101.5	101.2	0.35	29.45	87.2	120.4	249.3	0.972	1167
A-S-100-75	5-6-13	102.3	100.5	0.12	29.45	89.2	120.6	249.9	0.972	1167
A-S-100-80	5-6-13	97.0	100.0	0.03	29.45	91.2	121.9	249.9	0.971	1167
S-100	5-6-13	....	....	....	29.45	....	100.0	255.6	0.980	1171
A-S-80-10	4-25-13	106.9	81.9	1.32	29.35	89.4	110.5	249.3	0.977	1170
A-S-80-20	4-25-13	96.1	81.1	0.90	29.35	80.2	110.0	249.7	0.977	1170
A-S-80-22.3	4-21-13	104.8	82.4	0.83	29.41	83.1	114.2	248.2	0.973	1167
A-S-80-30	4-28-13	98.0	80.7	0.60	29.21	82.3	110.6	249.5	0.975	1170
A-S-80-50	4-28-13	99.5	80.7	0.32	29.21	84.3	109.9	250.1	0.976	1170
A-S-80-75	4-28-13	93.4	80.1	0.13	29.21	84.3	110.2	249.3	0.975	1169
A-S-80-80	4-30-13	91.0	77.1	0.04	29.37	87.5	110.1	251.1	0.975	1169
S-80	4-25-13	...	....	....	29.35	.	80.0	248.6	0.980	1167
A-S-60-10	4-15-13	118.7	61.6	0.99	29.20	85.1	113.9	254.6	0.977	1170
A-S-60-20	4-15-13	101.1	62.7	0.62	29.20	76.9	110.0	256.4	0.979	1171
A-S-60-30	4-15-13	106.4	61.8	0.46	29.20	82.2	111.3	255.5	0.978	1170
A-S-60-50	4-16-13	106.1	62.1	0.26	29.14	84.8	114.4	248.1	0.972	1166
A-S-60-75	4-16-13	106.3	61.0	0.11	29.14	87.2	109.9	247.3	0.972	1166
A-S-60-80	4-21-13	83.4	60.4	0.03	29.46	74.8	106.3	243.2	0.971	1165
S-60	4-21-13	..	....	....	29.46	....	60.0	231.4	0.978	1160

TABLE 11 (CONTINUED)

OBSERVED VALUES FOR ENGINE TESTS WITH AIR STEAM MIXTURES

	Condensate					i. h. p. Data					
	11	12	13	14	15	16	17	18	19	20	21
Designation	Temperature of Condensate (deg. F.)	Condensate Weighed (lb. per hr.)	Moisture Carried Away in Saturated Exhaust Air of Mixture (lb. per hr.)	Condensate in Separator Due to Heating Air (lb. per hr.)	Total Wet Steam (lb. per hr.)	m. e. p. Head End of Cylinder	m. e. p. Crank End of Cylinder	i. h. p. Head End of Cylinder	i. h. p. Crank End of Cylinder	Revolutions (per min.)	Net Weight on Scales
A-S-100-10	71.4	96.6	9.3	6.9	112.8	46.0	46.5	10.97	10.51	104.2	191
A-S-100-20	82.6	162.5	10.4	5.7	178.6	45.4	46.7	10.85	10.57	104.4	191
A-S-100-30	83.6	207.7	8.2	6.6	222.5	45.7	47.0	10.91	10.65	104.4	191
A-S-100-50	85.1	290.0	7.2	6.2	303.4	45.4	48.0	10.82	10.84	104.1	196
A-S-100-75	88.9	423.6	4.8	4.6	433.0	47.3	49.6	11.26	11.18	104.0	199
A-S-100-80	90.4	500.8	2.7	1.8	505.3	47.8	48.7	11.39	11.00	104.1	199
S-100	96.0	630.4	0.0	0.0	630.4	47.6	51.4	11.33	11.59	103.9	206
A-S-80-10	80.4	76.1	8.1	5.7	89.9	36.7	38.6	8.74	8.71	104.0	151
A-S-80-20	82.0	111.2	6.0	5.4	122.6	35.4	37.6	8.42	8.48	103.9	151
A-S-80-22.3	111.6	105.5	22.4	5.8	133.7	36.8	37.2	8.78	8.41	104.2	154
A-S-80-30	92.7	170.3	6.9	4.7	181.9	35.5	38.2	8.45	8.61	103.9	151
A-S-80-50	101.3	248.7	4.3	4.9	257.9	35.6	38.1	8.47	8.59	103.9	151
A-S-80-75	101.2	317.3	2.7	4.2	324.2	36.4	36.9	8.68	8.33	104.1	151
A-S-80-80	92.0	392.0	2.2	3.5	397.7	35.7	37.7	8.50	8.51	104.0	151
S-80	123.1	532.0	0.0	0.0	532.0	38.1	40.1	9.08	9.06	104.1	161
A-S-60-10	65.7	567.0	4.7	4.0	65.4	26.2	27.4	6.25	6.19	104.2	106
A-S-60-20	78.0	975.0	5.9	4.0	107.4	25.8	26.7	6.16	6.04	104.2	106
A-S-60-30	84.0	130.3	5.9	3.8	140.0	25.7	26.3	6.14	5.96	104.4	106
A-S-60-50	107.6	178.0	9.7	3.6	191.3	25.6	26.6	6.11	6.01	104.2	106
A-S-60-75	117.9	242.5	7.9	2.9	253.3	26.1	27.0	6.22	6.10	104.1	106
A-S-60-80	91.0	311.5	2.0	2.6	316.1	25.2	27.5	6.00	6.21	104.0	106
S-60	98.7	397.5	0.0	0.0	397.5	24.6	27.0	5.86	6.10	104.1	106

TABLE 12

OBSERVED VALUES AND COMPUTED RESULTS FOR ENGINE

73 74 76

Designation	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	Pressure at Throttle lb. per sq in. Gage)	Total Air		Total Steam, Terms of Dry Steam (Pres. in Item 1) lb. per hr.	Total Mixture lb. per hr. (Item 3 + Item 4)	Dry Steam in % of Mixture	Temperature (deg. F.)			i. h. p.	b. h. p.	Mech. Efficiency	cu. ft. Free Air per min. per i. h. p. (62° F., 14.7 lb. per sq. in. Abs.)	Air lb. per hr. per i. h. p.	Steam lb. per hr. per i. h. p.	Mixture lb. per hr. per i. h. p.	cu. ft. Free Air per min. per b. h. p. (62° F., 14.7 lb. per sq. in. Abs.)	Air lb. per hr. per b. h. p.	Steam lb. per hr. per b. h. p.	Mixture lb. per hr. per b. h. p.	Barometer (in. Mercury)	Room Temperature deg. F	B.t.u. Added per hr. by Steam = Heat Content of Steam in Item 4— Heat of Liquid at Exhaust Temp.	[illegible]
		cu. ft. Free Air per min. (60° F., 14.7 lb. per sq. in. Abs.)	lb. per hr.				Mixture at Throttle	In Steam Chest	At Exhaust															
A-S-100-10	100	144	654	112	766	14.6	234	231	128	21.48	19.90	92.6	6.65	30.3	5.2	35.5	7.16	32.7	5.6	38.3	29.3	81	122 010	45
A-S-100-20	100	114	522	177	701	25.2	268	266	158	21.42	20.00	93.2	5.32	24.3	8.3	32.6	5.72	26.1	8.9	35.0	29.3	83	157 990	45
A-S-100-30	100	98	444	220	664	33.1	284	282	171	21.56	20.00	92.9	4.51	20.6	10.2	30.8	4.87	22.2	11.0	33.2	29.3	86	230 850	45
A-S-100-50	100	73	334	296	630	47.0	302	300	185	21.66	20.44	94.5	3.37	15.5	13.6	29.1	3.57	16.3	14.5	30.8	29.5	78	306 500	45
A-S-100-75	100	43	197	423	620	68.2	321	319	201	22.44	20.70	92.2	1.92	8.8	18.8	27.6	2.08	9.5	20.4	29.9	29.5	78	431 500	45
A-S-100-80	100	23	105	494	599	82.5	330	329	207	22.39	20.74	92.8	1.03	4.7	22.0	26.7	1.11	5.1	23.8	28.9	29.5	80	501 190	45
A-S-100	100	0	0	622	. . .	100.0	338	338	212	22.92	21.40	93.4	0.00	0.0	27.1	. . . .	0.00	0.0	29.0	. . . .	29.5	81	627 500	
A-S-80-10	80	129	586	87	673	12.9	222	218	125	17.45	15.70	90.1	7.37	33.6	5.0	38.6	8.18	37.3	5.5	42.8	29.4	70	95 350	48
A-S-80-20	80	107	491	119	611	19.5	244	241	147	16.90	15.68	92.9	6.34	29.0	7.1	36.1	6.85	31.3	7.6	38.9	29.4	69	127 240	48
A-S-80-22.3	80	103	472	130	602	21.6	249	245	151	17.19	16.02	93.2	6.00	27.4	7.6	35.0	6.41	29.5	8.1	37.6	29.4	82	138 440	48
A-S-80-30	80	88	400	178	578	30.8	267	265	169	17.06	15.68	92.0	5.13	23.4	10.4	33.8	5.57	25.5	11.3	36.8	29.2	76	186 350	48
A-S-80-50	80	64	290	254	544	46.7	288	286	186	17.06	15.68	92.0	3.73	17.0	14.9	31.9	4.05	18.5	16.2	34.7	29.2	77	261 440	48
A-S-80-75	80	39	180	318	498	63.9	303	302	197	17.01	15.71	92.5	2.33	10.6	18.7	29.3	2.50	11.4	20.2	31.6	29.2	80	324 840	48
A-S-80-80	80	24	107	391	498	78.5	313	312	205	17.01	15.70	92.5	1.38	6.3	22.9	29.2	1.48	6.8	24.9	31.7	29.4	85	395 [illegible]	48
A-S-80	80	0	0	525	. . .	100.0	326	325	212	18.14	16.75	92.5	0.00	0.0	28.9	. . . .	0.00	0.0	31.3	. . . .	29.4	70	527 800	
A-S-60-10	60	102	448	63	511	12.3	201	198	121	12.44	11.05	89.0	7.90	36.1	5.0	41.1	8.88	40.5	5.7	46.2	29.2	75	69 050	51
A-S-60-20	60	80	367	105	472	22.2	235	233	151	12.20	11.05	90.7	6.58	30.1	8.6	38.7	7.28	33.2	9.5	42.7	29.2	73	111 150	51
A-S-60-30	60	67	307	138	445	31.0	251	248	164	12.10	11.08	91.6	5.57	25.5	11.4	36.9	6.10	27.8	12.4	30.2	29.2	75	144 230	51
A-S-60-50	60	52	236	189	425	44.5	271	269	182	12.12	11.05	91.3	4.26	19.5	15.6	35.1	4.69	21.3	17.1	38.4	29.1	79	194 400	51
A-S-60-75	60	32	146	250	396	63.1	287	285	196	12.32	11.03	89.6	2.61	11.8	20.3	32.1	2.91	13.3	22.6	35.9	29.1	83	253 810	51
A-S-60-80	60	19	82	311	393	79.1	298	297	205	12.21	11.02	90.4	1.46	6.7	25.5	32.2	1.65	7.5	28.2	35.7	29.5	76	313 720	51
A-S-60	60	0	0	390	. . .	100.0	309	307	212	11.96	11.03	92.4	0.00	0.0	32.6	. . . .	. . . .	0.0	35.3	. . . .	29.5	79	390 600	

TABLE 12

TED RESULTS FOR ENGINE TESTS WITH AIR-STEAM MIXTURES

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21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	
Barometer in Mercury	Room Temperature deg. F.	B. t. u. Added per hr. by Steam = Heat Content of Steam in Item 4—Heat of Liquid at Exhaust Temp.	lb. Air Req. per i. h. p. hr. at 100° F., and Pressure in Item 1	i. h. p. hr. Obtainable from Air (Item 3) 100° F. Pressure in Item 1 (Item 3 ÷ Item 24)	i. h. p. hr. Added by Steam (Item 10 Less Item 25)	B. t. u. Req. per i. h. p. hr. Added by Steam (Item 23 ÷ Item 26)	lb. Dry Steam Used per i. h. p. hr. Added (Item 4 ÷ Item 26)	Engine Water-Rate Pressure in Item 1 25% Cut-off	i. h. p. hr. Obtainable from Steam (Item 4) (Item 4 ÷ Item 29)	i h p. hr. Obtainable Using Air and Steam Separately (Item 25 + Item 30)	i. h. p. Gained by Mixing (Item 10 Less Item 31)	Per Cent Gain (Item 32 ÷ Item 31)	lb. Air Req. per b. h. p. hr. at 100° F. and Pressure in Item 1	b. h. p. hr. Obtainable from Air (Item 3) 100° F. Pressure in Item 1 (Item 3 ÷ Item 24)	b. h. p. Added by Steam (Item 11 Less Item 35)	B. t. u. Req. per b. h p hr. Added by Steam (Item 23 ÷ Item 36)	lb. Dry Steam Used per b. h p. hr. Added (Item 4 ÷ Item 36)	Engine Water-Rate Pressure in Item 1 25 % Cut-off	b. h. p. hr. Obtainable from Steam (Item 4) (Item 4 ÷ Item 39)	b. h. p. hr. Obtainable Using Air and Steam Separately (Item 35 + Item 40)	b. h. p. Gained by Mixing (Item 11 Less Item 41)	Per Cent Gain (Item 42 ÷ Item 41	Designation
29 3	81	122 010	45.3	14.36	7.12	17 120	15.7	27.1	4.14	18.50	2.98	16.1	54.4	11.97	7.93	15 390	14.1	29.0	3.87	15.84	4.06	25.6	A-S-100-10
29 3	83	187 930	45.3	11.50	9.92	18 940	17.8	27.1	6.55	18.05	3.37	18.7	54.4	9.60	10.40	18 050	17.0	29.0	6.11	15.71	4.29	27 2	A-S-100-20
29 3	86	230 850	45.3	9 81	11 75	19 660	18.7	27.1	8.13	17.94	3.62	20.2	54 4	8.18	11.82	19 520	18.6	29.0	7.60	15.78	4.22	26 7	A-S-100-30
29 5	78	306 500	45.3	7.38	14 28	21 410	20 7	27.1	10.92	18.30	3.36	18.3	54.4	6.15	14.29	21 430	20.7	29.0	10.20	16.35	4.09	25.0	A-S-100-50
29 5	78	431 500	45.3	4.35	18.09	23 810	23.4	27.1	15.62	19.97	2.47	12.4	54.4	3.62	17.08	25 250	24.8	29.0	14.60	18.22	2.48	13.6	A-S-100-75
29 5	80	501 190	45.3	2.32	20 07	24 980	24.6	27.1	18.23	20.55	1.84	9.0	54 4	1.93	18.81	26 620	26.2	29.0	17.05	18.98	1.76	9.3	A-S-100-80
29.5	81	627 500	....	....	22.92	27 230	27.1	27.1	....	....	....	....	....	....	21.40	29 300	29.0	29.0	...	....	....	. .	S-100
29 4	70	95 350	48.0	12.19	5.26	18 100	16.5	28.9	3.01	15.20	2.25	14.8	60.8	9.64	6.06	15 700	14.3	31.3	2.78	12.42	3.28	26 4	A-S-80-10
29 4	69	127 240	48.0	10.19	6.71	19 020	17.7	28.9	4.12	14.31	2 59	18.1	60.8	8.09	7.59	16 780	15.7	31.3	3.81	11.90	3.78	31.8	A-S-80-20
29.4	82	138 440	48.0	9.83	7.36	18 780	17.7	28.9	4.50	14.33	2.86	19.9	60.8	7.78	8.24	16 800	15.8	31.3	4.16	11.94	4.08	34.2	A-S-80-22 5
29 2	76	186 350	48.0	8.32	8.74	21 310	20.4	28.9	6.16	14.48	2.58	17.8	60.8	6.59	9.09	20 500	19.6	31.3	5.70	12.29	3.39	27.6	A-S-80-30
29 2	77	261 440	48.0	6.04	11.02	23 740	23.0	28.9	8.80	14.84	2.22	14.9	60.8	4.79	10.89	24 000	23.3	31.3	8.12	12.91	2.77	21.4	A-S-80-50
29 2	80	324 840	48.0	3.76	13.25	24 430	24.0	28.9	11.00	14.76	2.25	15.2	60 8	2.97	12.74	25 470	24.9	31.3	10.15	13.12	2.59	19.7	A-S-80-75
29 4	85	395 690	48.0	2.24	14.77	26 750	26.5	28.9	13.53	15.77	1.24	7.9	60.8	1.76	13.94	28 370	28.0	31.3	12.50	14.26	1 44	10.1	A-S-80-80
29.4	70	527 800	....	....	18.14	28 950	28.9	28.9	....	....	....	....	....	....	16.75	31 500	31.3	31.3	....	....	....	.	S-80
29.2	75	69 050	51.1	8.77	3.67	18 790	17.1	32 6	1.93	10.70	1.74	16.3	73 1	6.14	4.91	14 030	12.8	35.3	1.78	7.92	3.13	39.6	A-S-60-10
29.2	73	111 150	51.1	7.17	5.03	22 080	20.9	32.6	3.22	10.39	1.81	17.4	73.1	5.02	6.03	18 460	17.4	35.3	2.98	8.00	3.05	38.1	A-S-60-20
29 2	75	144 230	51.1	6.02	6.08	23 710	22.7	32.6	4.24	10.26	1.84	17.9	73.1	4.21	6.87	21 000	20.1	35.3	3.91	8.12	2.96	36 4	A-S-60-30
29.1	79	194 400	51.1	4.61	7.51	25 770	25.2	32.6	5.81	10.42	1.70	16 3	73.1	3.23	7.82	24 850	24.2	35.3	5.35	8.58	2 47	28 8	A-S-60-50
29 1	83	253 810	51.1	2.86	9.46	26 790	26.4	32 6	7.68	10.54	1 78	16.9	73 1	2.00	9.03	28 100	27.7	35.3	7.08	9.08	1.95	21 5	A-S-60-75
29.5	76	313 720	51.1	1.61	10.60	29 580	29.4	32 6	9.56	11.17	1 04	9.3	73.1	1.13	9.89	31 720	31.4	35.3	8.81	9.94	1.08	10 9	A-S-60-80
29.5	79	390 600	....	....	11.96	32 540	32.6	32.6	....	....	....	....	....	....	11.03	35 400	35.3	35.3	....	....	. .	.	S-60

from the predetermined ratio as shown by the last number of the test designation. This deviation, however, caused no difficulty. Items 7 and 8 show that the temperature difference between the throttle and the steam chest is small compared with that found when air alone is used. Even a small quantity of steam, 10 to 15 per cent of the mixture, will make this difference 4 degrees Fahrenheit or less; in the air tests at high temperatures this difference was often more than 40 degrees.

Item 6, the per cent of steam in the mixture, plotted against Item 7, the temperature at the throttle, for each total pressure gives the curves shown in Fig. 17. These curves suggest a method for regulating the proportions of the mixture without weighing the condensate. With a given cut-off and brake load, and for a specified total pressure, the valves in the air and steam lines can be adjusted to give the temperature at the throttle which results from the desired mixture.

In these air-steam tests the exhaust temperature, Item 9, is reliable and probably in no case is it more than 1 or 2 degrees in error. In all these tests it was considerably above the temperature of the atmosphere and surrounding objects; when steam alone was used, exhausting at atmospheric pressure, the thermometer read 212 degrees Fahrenheit. These exhaust temperatures show that the steam contained in an air-steam mixture is used more efficiently than it would be in a non-condensing steam plant; and, in the case of mixtures having small percentages of steam, it is used with efficiencies comparable to those obtained with condensing engines. In test A-S-100–10 the exhaust temperature was 128 degrees Fahrenheit, which is the temperature that would result in a condensing plant maintaining a 25 inch vacuum. The exhaust temperature may be regarded as a measure of the condenser effect produced from the use of mixtures of air and steam, resulting from the fact that the total exhaust pressure is the sum of the partial pressures of the steam and air.

When even a small quantity of steam was admitted to the cylinder with the air the difficulties with lubrication and the excessive internal friction were overcome, as shown by the indicated and brake horse power, Items 10 and 11. The difference between these items gives a friction horse power of about 1.5 for 100 pounds, 1.4 for 80 pounds, and 1.2 for 60 pounds. In the air tests with serial numbers 22-31 and 48-51, in which the air was 400 degrees at the throttle, about the same indicated horse power was developed as in the air-steam tests at 100

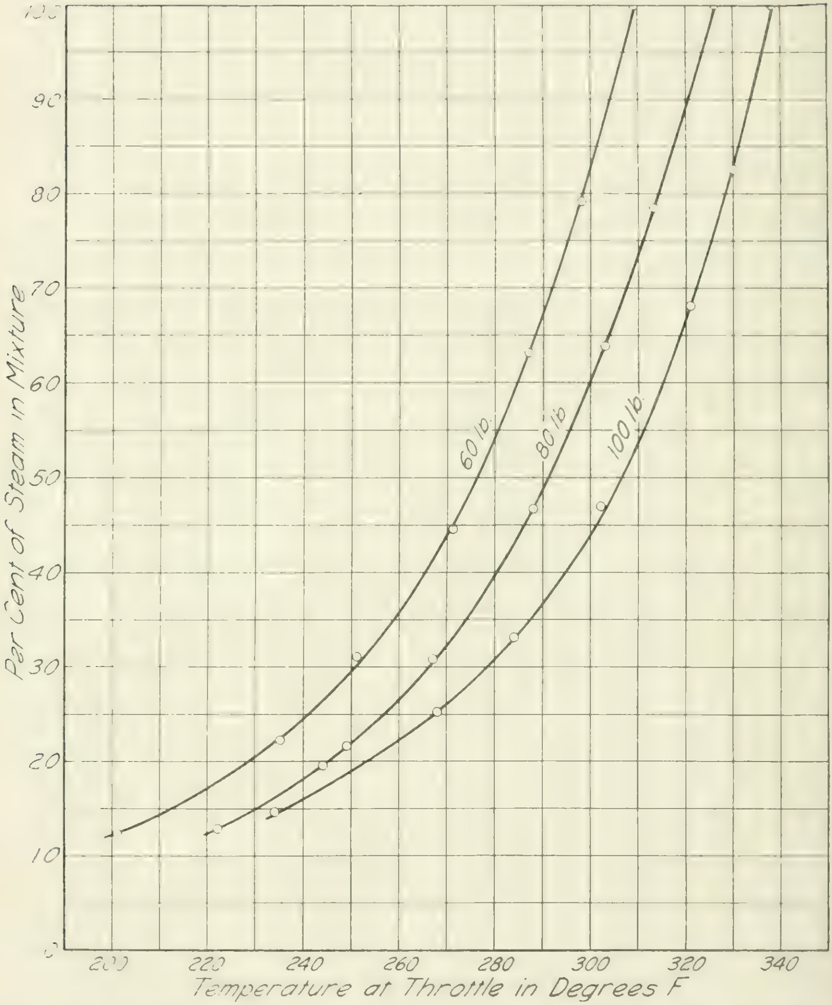

Fig. 17. Relation between Temperatures and Pressures at Throttle and Percentages of Steam in Air Steam Mixture

pounds gage. In the air tests, however, the friction horse power was 1.7, showing that even with 100 degrees at the throttle the internal friction was not quite reduced to its minimum value, which, when the engine was developing 21 indicated horse power with 100 pounds pressure at the throttle, may be taken as 1.5 horse power.

Item 12 shows that when the load is nearly constant the mechanical efficiency does not vary with the per cent of steam in the mixture. When the initial pressure is decreased and the cut-off kept constant there is a slight decrease in the efficiency due to decrease of the load, as follows:

Pressure at Trottle (gage)	Average i.h.p.	Average Mechanical Efficiency
100	21.98	93.1
80	17.23	92.2
60	12.19	90.7

The air, steam, and mixture rate per indicated horse power given in Items 14, 15, and 16 are plotted to per cent of mixture in Figures 18, 19, 20. In each figure, curve (2) represents the air weight, curve (3) the steam weight, and curve (1) the weight of the mixture. Curve (1) is therefore the summation curve of curves (2) and (3). Figure 21 is the curve (1) of Figures 18, 19, and 20 drawn to a larger scale. It is unnecessary to plot the corresponding curves for brake horse power since the air and steam consumption per brake horse power per hour may be obtained from these curves by dividing particular values for a given pressure by the average mechanical efficiency for that pressure as shown above.

22. *Economic Results.*—The presentation of the economic effects resulting from mixing air and steam begins with Item 23, which gives the total heat above the exhaust temperature of the equivalent steam supplied during the test. This is the total heat chargeable to the engine, since the condensed exhaust steam is assumed to be used as boiler feed water. In Item 25 the indicated horse power per hour obtainable from the air in the mixture if used alone at 100 degrees Fahrenheit is found by dividing Item 3 by Item 24, which is the air-rate per indicated horse power at the given throttle pressure and 100 degrees Fahrenheit determined from air tests. Subtracting Item 25 from Item 10 gives the indicated horse power which may be assumed to

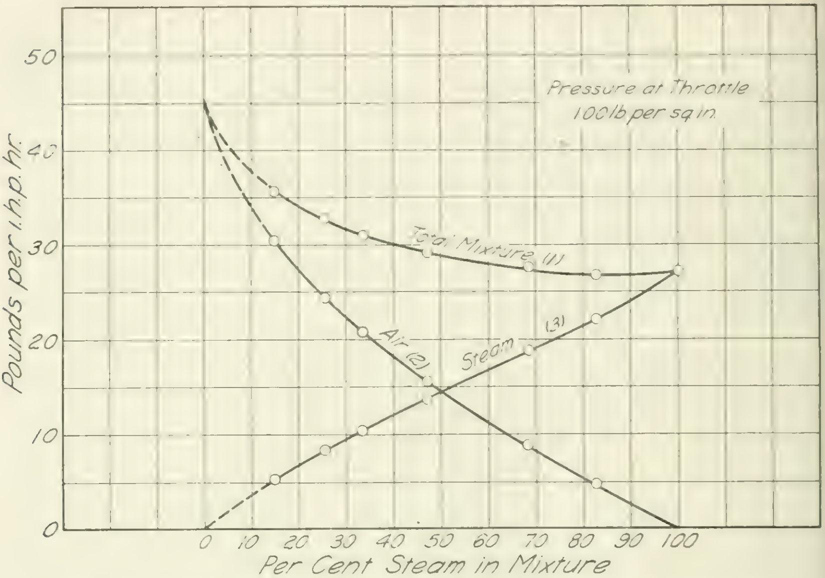

FIG. 18. CONSUMPTION OF AIR, STEAM, AND MIXTURE PER INDICATED HORSE POWER PER HOUR FOR DIFFERENT PERCENTAGES OF STEAM AND MIXTURE; THROTTLE PRESSURE 100 LB. PER SQ. IN.

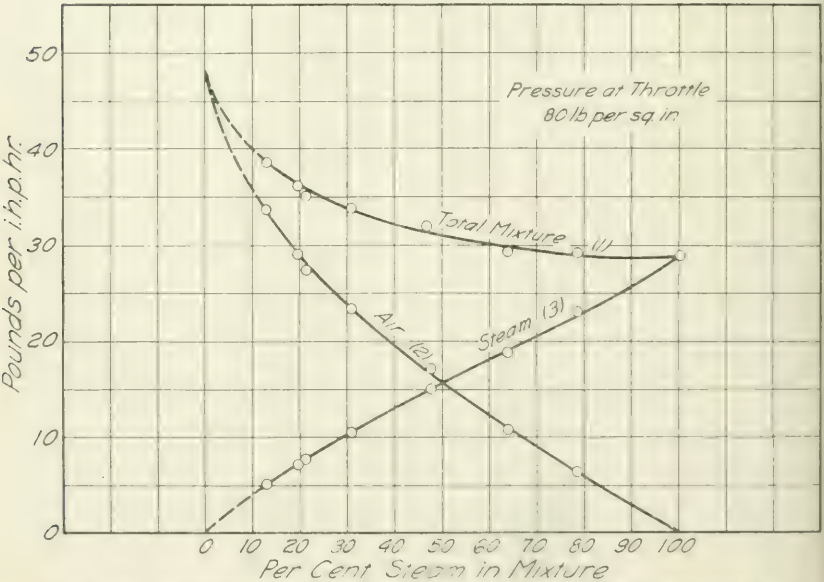

FIG. 19. CONSUMPTION OF AIR, STEAM, AND MIXTURE PER INDICATED HORSE POWER PER HOUR FOR DIFFERENT PERCENTAGES OF STEAM AND MIXTURE; THROTTLE PRESSURE 80 LB. PER SQ. IN.

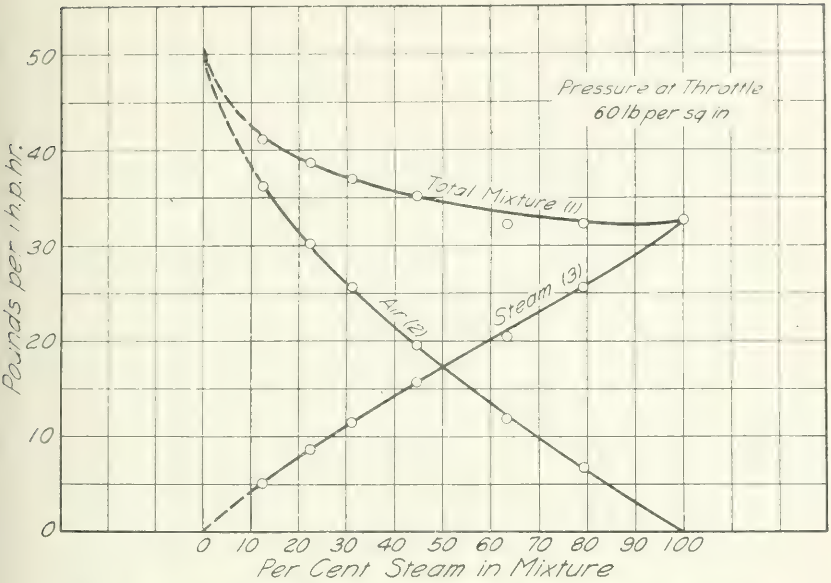

FIG. 20. CONSUMPTION OF AIR, STEAM, AND MIXTURE PER INDICATED HORSE POWER PER HOUR FOR DIFFERENT PERCENTAGES OF STEAM AND MIXTURE; THROTTLE PRESSURE 60 LB. PER SQ. IN.

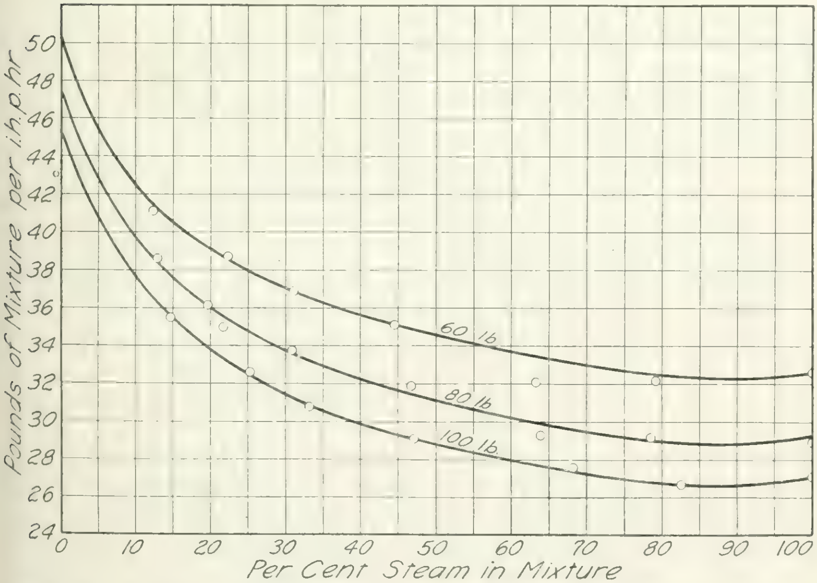

FIG. 21. CONSUMPTION OF AIR, STEAM, AND MIXTURE PER INDICATED HORSE POWER PER HOUR FOR DIFFERENT PERCENTAGES OF STEAM AND MIXTURE

be chargeable to the steam in the mixture. Items 27 and 28 give the cost of the added horse power in terms of British thermal units and pounds of dry steam. These two items increase steadily as the per cent of steam is increased. With 15 per cent of steam in the mixture the number of British thermal units required per indicated horse power per hour added is slightly more than one-half the number required per indicated horse power per hour when steam alone is used. Again, comparing the tests with 15 per cent steam with air tests at the same pressure, it is seen that the number of British thermal units required per indicated horse power per hour added in the case of air-steam mixtures is about twice as great as the number required per indicated horse power per hour gained by reheating air alone. Figure 22 shows the characteristics here discussed.

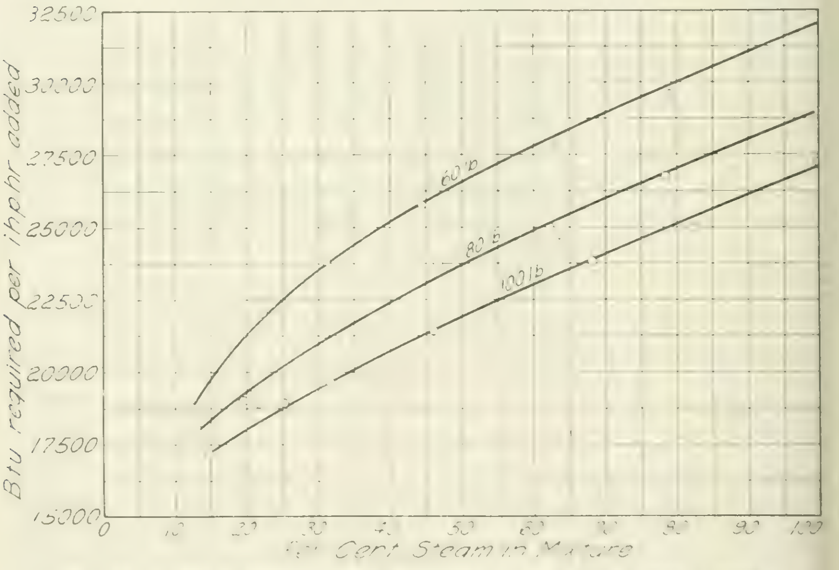

Fig. 22. Heat Required per Additional Indicated Horse Power per Hour for Different Throttle Pressures and Different Percentages of Steam in Mixture

Item 29 shows the best obtainable steam consumption in pounds per indicated horse power per hour, with cut-off at 25 per cent of the stroke and an initial pressure equal to the total initial pressure of the mixture in a particular test, as determined in a series of tests using steam alone. If the engine had been operated at full load under the conditions specified in Item 29 until the weight of steam given in Item 4 was used, the number of indicated horse power hours shown in Item 30 would have been obtained. By a similar process the number of indicated horse power hours which could have been secured with the actual weight of air used in a particular test is given in Item 25. Therefore the sum of Items 25 and 30, Item 31, represents the power which could have been realized in the engine tested if the air and steam in the mixture had been used separately. The actual power secured from the mixture of air and steam in a particular test is shown under Item 10. Hence the difference between Items 10 and 31, Item 32, is the gain in power, in indicated horse power hours, which results when the engine is operated with the given mixture of the air and steam. Through this method of reasoning it is evident that the mixture of the air and steam affords a thermodynamic advantage over the use of the separate ingredients amounting to as much as 20.2 per cent, based on indicated horse power, (Test A–S–100–30), or 39.6 per cent based on brake horse power, (Test A–S–60–10).

By a similar process of calculation the economic results of the tests, based on brake horse power, are calculated and presented in Table 12 under Items 34 to 43 inclusive.

The astonishing results of these tests of air-steam mixtures must not be construed as an argument for the use of such a medium in the generation of power. If the air is compressed by a steam driven compressor the total steam consumption per horse power produced by an engine using a mixture of air and steam would probably be greater than through the use of steam alone, when the steam used in the compressor is included. When, however, a supply of compressed air is available and is used for generating power, and steam is also available, the mixture of the two affords a distinct advantage over the use of air alone or of steam alone if the cost of compressing the air is disregarded.

VI. GENERAL SUMMARY AND CONCLUSIONS

23. *Reheaters and Reheating.*—

(a) Commercial types of small external combustion reheaters will under favorable conditions and when operated at their best capacity supply 60 per cent of the heat in the fuel to the compressed air.

(b) The efficiency is dependent upon the design of the heating surface and its relation to the furnace or fire-pot. The maximum efficiency in improperly designed apparatus may be very low. Heating surface in direct contact with the fire-pot is three times as effective as such surface not in direct contact with the fire or very close to it.

(c) Internal combustion reheaters may attain efficiencies of 80 per cent or more when operating at their best capacity with reheated air temperatures of approximately 400 degrees Fahrenheit.

(d) The efficiencies of all types of reheaters decrease rapidly when the reheater is operated at low capacities. The size of the reheater should therefore be carefully selected with reference to the average capacity demanded by the service.

(e) The results of this investigation indicate that a chimney draft of from 0.15 to 0.25 inches of water is essential for the best results.

(f) One thousand pounds of air per hour or 219 cubic feet of free air per minute can be heated through a range of 100 degrees Fahrenheit with $3\frac{1}{4}$ to $3\frac{1}{2}$ pounds of fuel whose calorific value is 13 000 British thermal units per pound.

(g) To avoid excessive losses of heat the reheater must be installed as close as possible to the motor it serves.

24. *Expansive Use of Reheated Air.*—

(a) With the particular engine employed in this investigation operating with a cut-off at one-quarter stroke the air consumption per indicated horse power per hour decreases as follows, as the air temperature rises from 100 degrees to 400 degrees Fahrenheit:

With an initial gage pressure of 100 pounds per square inch the air consumption varies from approximately 45 pounds to 33 pounds, or from 10 cubic feet to 7.3 cubic feet of free air per minute.

With an initial gage pressure of 80 pounds per square inch the air consumption varies from approximately 48 pounds to 35.23 pounds or 10.5 cubic feet to 7.73 cubic feet of free air per minute.

With an initial gage pressure of 60 pounds per square inch the air consumption varies from approximately 51.1 pounds to 38.3 pounds, or 11.1 to 8.45 cubic feet of free air per minute.

(b) Similarly the air consumption per brake horse power per hour with other conditions as given above varies as follows:

at 100 pounds gage pressure, from 58.3 pounds to 36.2 pounds per hour, or from 12.77 cubic feet to 7.93 cubic feet of free air per minute;

at 80 pounds gage pressure, from 63.35 pounds to 39.45 pounds per hour, or from 13.9 cubic feet to 8.65 cubic feet of free air per minute;

at 60 pounds gage pressure, from 76.43 pounds to 45.84 pounds per hour, or 16.7 cubic feet to 10 cubic feet of free air per minute.

(c) Air at 400 degrees Fahrenheit will yield the same brake power with two-thirds the compressor capacity and steam consumption required when the air is used at 100 degrees Fahrenheit.

(d) When air is reheated from 100 degrees to 400 degrees Fahrenheit an indicated horse power per hour is gained by the expenditure of approximately 8700 to 9300 British thermal units, or 1.15 pounds of fuel, whose heating value is 13 000 British thermal units, burned in the reheater with 60 per cent efficiency. The heat required for the first 100 degree rise in temperature is particularly effective. In this range an indicated horse power was gained by the addition of about 7700 British thermal units or less than one pound of fuel.

(e) Reheating the air from 100 degrees to 400 degrees Fahrenheit permits the gain of a brake horse power per hour with an expenditure of heat as low as 8500 British thermal units. In Test 3-100-200-15 a brake horse power was gained with an expenditure of only 7035 British thermal units or about 0.9 pounds of fuel. Test 44–60–200–25 shows an even more astonishing result for a rise of temperature of 100 degrees only.

(f) For the initial pressures employed in these tests the highest efficiencies were obtained when the cut-off was at about one-quarter of the stroke. Complete expansion does not show equally good results.

(g) The use of reheated air leads to an improvement in lubrication, an astonishing increase in mechanical efficiency, and the prevention of ice formation in motors using air expansively. In practice the chief reason for the use of reheaters is the prevention of interference with the operation of the motor through the accumulation of ice in the exhaust pipe.

25. *Use of Steam for Reheating Air.*—

(a) Steam is a convenient medium for reheating air, and a mixture of air and steam containing as little as 10 per cent of steam will entirely prevent the operating difficulties experienced when cold air is used in expansion motors through defective lubrication and the formation of ice in the exhaust pipe.

(b) A greater amount of work is obtained through the use of a mixture of air and steam than could be realized by the separate use of the same weights of each of the two substances. When the steam constitutes 15 to 20 per cent of the mixture there is a gain of from 16 to 20 per cent in the indicated horse power, and of from 25 to 35 per cent in the brake horse power of the engine. An increase in the weight of steam in the mixture reduces the gain.

(c) A much greater expenditure of heat is required per horse power gained when steam is used than is necessary with other methods of reheating.

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UNIVERSITY OF ILLINOIS
ENGINEERING EXPERIMENT STATION

BULLETIN No. 131 JULY, 1922

A STUDY OF AIR-STEAM MIXTURES

BY
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WITH
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A STUDY OF AIR-STEAM MIXTURES

I. Introduction

1. *Purpose of Investigation.*—This investigation of air-steam mixtures is the outgrowth of an investigation of the reheating of compressed air by C. R. Richards, Director of the Engineering Experiment Station and Dean of the College of Engineering, and J. N. Vedder, Research Assistant in Mechanical Engineering, the results of which have been presented in Engineering Experiment Station Bulletin No. 130. In this earlier investigation the employment of steam as a reheating agent was found to result in an increased thermal efficiency for the mixture of air and steam, as compared with the use of either air or steam separately, in an engine operating expansively. These results were such as to justify a further study of air-steam mixtures.

It is the purpose of this bulletin to treat the subject of air-steam mixtures in considerable detail both by means of a theoretical discussion and by reporting actual tests made with different proportions of air and steam at different initial air temperatures and under various load conditions. To facilitate the practical application of the results secured, the thermal properties of various mixtures are presented in the form of Mollier charts. These charts will greatly simplify the solution of problems connected with the subject.

2. *Acknowledgments.*—This investigation was conducted by Leroy A. Wilson,[*] Research Associate in Mechanical Engineering, under the general direction of Dean C. R. Richards. It was due to Wilson's painstaking efforts that the investigation was brought to a successful conclusion. Valuable assistance in the experimental work was rendered by L. C. Lichty and W. P. Lukens, Research Fellows in the Engineering Experiment Station; and in the preparation of the Mollier diagrams for air-steam mixtures by John A. Dent,[†]

---

[*] Now Professor of Mechanical Engineering at the Oklahoma Agricultural and Mechanical College, Stillwater.

[†] Now Associate Professor of Mechanical Engineering at the University of Kansas, Lawrence.

Associate in Mechanical Engineering. The section on the thermodynamics of air-steam mixtures has been carefully revised by G. A. Goodenough, Professor of Thermodynamics, and it is, therefore, an authoritative discussion of the subject.

II. TESTS WITH STEAM, AIR, AND AIR STEAM MIXTURES

3. *Purpose of Tests.*—The purpose in operating a reciprocating engine with a working medium composed of a mixture of air and steam was to determine the possible advantages in thermal efficiency resulting from the use of such mixtures, having different proportions of the two components and with different initial temperatures of the compressed air. As a basis for the comparison of results it was necessary to determine the engine economy when using steam alone and air alone at various initial temperatures.

4. *Tests with Steam.*—The tests with steam were made to determine the steam consumption at different loads. A gage pressure of 100 pounds per square inch was used. The quality of the steam was determined by means of a throttling calorimeter located just above the throttle valve. All observations and indicator diagrams were taken every 5 minutes during 30-minute tests. The following data were recorded:

(1) Net brake load in pounds.
(2) Revolutions per minute, by the revolution counter.
(3) Pressure at throttle (kept constant) in pounds per square inch.
(4) Temperature at throttle in degrees Fahrenheit.
(5) Temperature in steam chest in degrees Fahrenheit.
(6) Temperature of engine room in degrees Fahrenheit.
(7) Temperature in exhaust pipe in degrees Fahrenheit.
(8) Pressure in exhaust pipe in pounds per square inch.
(9) Temperature in calorimeter in degrees Fahrenheit.
(10) Temperature of condensed steam in degrees Fahrenheit.
(11) Weight of condensed steam in pounds.
(12) Barometer height in inches of mercury.

The results of the steam tests are given in Table 1 and are represented graphically by the curves in Fig. 1. For convenience of reference the tests are designated in the tables by the letter S, followed by the gage pressure at the throttle. The lowest steam consumption per indicated horse power per hour attained in this series of tests was 28.9 pounds; in similar tests run by J. N. Vedder and reported in Bulletin No. 130 the lowest consumption was 27.1 pounds.

TABLE 1

RESULTS OF TESTS USING STEAM ALONE

	Miscellaneous Data										Steam Data	
	1	2	3	4	5	6	7	8	9	10	11	12
Designation	Date	Pressure at Throttle Valve lb. per sq. in. gage	Temp. at Throttle Valve deg. F	Temp. in Steam Chest deg. F.	Temp. of Engine Room deg. F.	Pressure in Exhaust Pipe lb. per sq in. gage	Temp in Exhaust Pipe deg. F.	Barometer in Hg	Atmospheric Pressure lb per sq. in.	Pressure Calorimeter lb per sq. in. gage	Temp. Calorimeter deg. F.	Quality
S-100	12-11-14	100	338	338.0	80	0.10	210.5	29.63	14.5	101.5	282	0.994
	12-28-14	100	337	337.7	84	0.00	209.5	29.32	14.4	101.5	284	0.995
S-150	12-11-14	100	338	337.7	86	0.30	210.5	29.62	14.5	101.5	281	0.993
S-175	12-11-14	100	338	337.7	78		211.0	29.63	14.5	101.5	281	0.993
	12-28-14	100	337	337.7	80	0.10	210.5	29.32	14.4	101.5	285	0.995
	2-17-15	100	338	337.7	82	0.20		29.62	14.5	101.5	281	0.993
S-200	12-11-14	100	338	337.7	81	0.40	211.8	29.63	14.5	101.5	281	0.993
	2-17-15	100	338	337.7	83	0.25		29.62	14.5	101.5	281	0.993
S-225	12-11-14	100	338	337.7	80		212.5	29.63	14.5	101.5	281	0.993
	12-28-14	100	337	337.7	85	0.30	212.0	29.32	14.1	100.0	275	0.991
	2-17-15	100	338	337.7	83	0.35		29.62	14.5	101.5	281	0.993
S-250	12-11-14	100	338	337.7	84		213.0	29.63	14.5	101.5	280	0.993
	12-28-14	100	337	337.7	83	0.35	213.0	29.32	14.4	101.5	285	0.995

TABLE 1 (CONTINUED)

RESULTS OF TESTS USING STEAM ALONE

Designation	Steam Data (Cont'd)			Engine Data								Results	
	13	14	15	16	17	18	19	20	21	22	23	24	25
	Heat Content B. t. u. per lb.	Weight Wet Steam lb. per 30 min.	Weight Dry Steam lb. per hr.	r. p. m.	m. e. p. Head End	m. e. p. Crank End	i. h. p. Head End	i. h. p. Crank End	Total i. h. p.	b. h. p.	Mechanical Efficiency	Dry Steam per i. h. p. hr. lb.	Dry Steam per b. h. p. hr. lb.
S–100	1185.5	211.3	420	107.00	25.41	25.24	6.23	5.89	12.12	10.70	0.883	34.6	39.2
	1186.5	213.7	425	107.77	24.83	27.33	6.12	6.38	12.50	10.78	0.862	34.0	39.1
S–150	1185.0	263.6	523	105.47	36.29	36.40	8.77	8.32	17.09	15.82	0.925	30.6	33.0
S–175	1185.0	295.2	586	104.80	41.69	42.27	10.02	9.60	19.62	18.34	0.935	29.9	32.0
	1186.5	300.8	598	105.97	41.27	44.16	10.02	10.13	29.15	18.54	0.920	29.7	32.2
	1185.0	296.4	589	104.54	41.29	43.46	9.89	9.84	19.73	18.29	0.927	29.8	32.2
S–200	1185.0	329.3	654	104.20	46.74	47.97	11.16	10.83	21.99	20.84	0.948	29.7	31.4
	1185.0	328.8	653	104.00	47.06	48.83	11.21	11.00	22.21	20.80	0.936	29.4	31.4
S–225	1185.0	363.2	721	103.57	53.24	53.61	12.64	12.02	24.66	23.30	0.945	29.2	31.0
	1182.5	365.3	724	104.73	52.41	53.56	12.58	12.16	24.74	23.56	0.951	29.3	30.8
	1185.0	367.2	729	103.30	52.00	54.56	12.30	12.21	24.51	23.24	0.948	29.7	31.3
S–250	1184.5	399.5	793	103.03	57.75	58.56	13.63	13.06	26.69	25.76	0.964	29.7	30.8
	1186.5	404.0	804	104.30	59.36	60.36	14.18	16.62	27.80	26.08	0.938	28.9	30.8

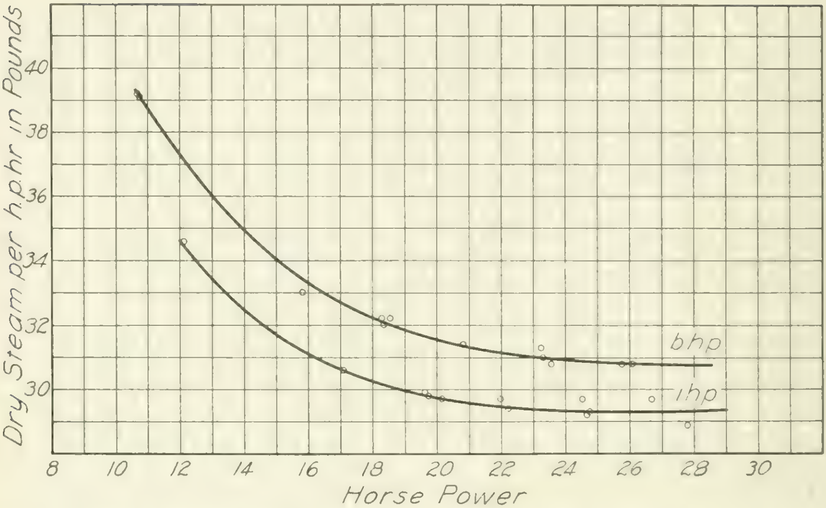

FIG. 1. STEAM CONSUMPTION PER HORSE POWER PER HOUR, USING STEAM ALONE

5. *Tests with Air.*—The tests with air were made to determine the air consumption at different loads for different initial air temperatures. The water vapor in the air froze in all tests made with air which had a temperature of 300 degrees Fahrenheit or less. In every test in which the exhaust temperature was below 32 degrees Fahrenheit the exhaust valve clogged with ice. When the back pressure caused by the stoppage became appreciable the test was terminated.

All tests were started with the engine comparatively warm, and only data and indicator diagrams secured during the last thirty minutes of a test were used. This practice was adopted to obtain data representative of normal running conditions. It was found, however, that owing to increasing friction in the engine caused by the formation of ice and from less effective lubrication, the mechanical efficiency decreased as the test progressed. It was impossible to get satisfactory equilibrium conditions with air alone except by heating the air to an initial temperature sufficient to prevent the exhaust temperature falling below 32 degrees Fahrenheit. By taking indicator diagrams and Venturi meter data simultaneously at stated intervals, under the conditions named, and computing the air consumption per indicated horse power per hour, this was found to remain practically constant throughout the test. The air consumption per brake horse power per hour varied for the reasons already stated; therefore, for these tests it has been averaged for the 30-minute period.

An unsuccessful attempt was made to overcome lubrication difficulties by using alcohol and graphite. A mixture of gas-engine oil and graphite seemed to give the best lubrication results. Were it practicable to free the air from moisture before admitting it to a reciprocating engine, there would probably be little mechanical difficulty in using it expansively, but with such large quantities of air as were used in these tests, no convenient nor simple means for drying the air could be devised.

The tests with air were made at several loads with an initial gage pressure of 100 pounds per square inch at the throttle, and with initial temperatures of approximately 100, 200, 300, and 400 degrees Fahrenheit, respectively, at the entrance to the mixing chamber. When air was used the engine would not carry over-loads satisfactorily, probably because of the small size of the supply pipe and the consequent drop in pressure accompanying a late cut-off. The conditions of operation were such as to prevent the formation of a loop at the toe of the indicator diagram; hence complete expansion to back pressure determined the limit of earliest cut-off.

The condenser was disconnected during the air tests, the air being exhausted directly into the room. A pentane thermometer reading in degrees centigrade was placed in a mercury well as near the cylinder as possible to indicate the exhaust temperature. Owing to the proximity of steam pipes and the heat of the surrounding atmosphere the lower temperatures recorded are probably incorrect. It should also be noted that there was a tendency for snow and ice to accumulate in the exhaust pipe around the thermometer cup, but by pounding occasionally on the outside of the pipe these obstructions were dislodged.

The air pressure was set at 100 pounds per square inch gage and kept constant by regulating the pressure of the steam operating the compressor. A preliminary run of two or three hours duration was necessary to heat the pipes and enable adjustments of apparatus, particularly of the reheater, to be made; during the run sufficient steam was allowed to mix with the air to prevent freezing. Between runs it was necessary to pass steam through the engine to thaw the

TABLE 2

RESULTS OF TESTS USING AIR ALONE

Designation	Miscellaneous Data							Air Data					
	1	2	3	4	5	6	7	8	9	10	11	12	13
	Date	Pressure at Throttle Valve lb. per sq. in. gage	Temp. at Throttle Valve deg. F.	Temp. in Steam Chest deg. F.	Temp. of Engine Room deg. F.	Pressure in Exhaust Pipe lb. per sq. in. gage	Temp. in Exhaust Pipe deg. F.	Pressure at Venturi Meter lb. per sq. in. gage	Temp. at Venturi Meter deg. F.	Venturi Head by Manometer in. Hg.	Temp. Entering Mixer deg. F.	Barometer in. Hg.	Atmospheric Pressure lb. per sq. in.
A-100-125	12-30-14	100	101	95	69	0.10	—43.0	102.20	125.1	2.176	101	29.63	14.50
A-100-150	12-30-14	100	102	98	68	0.10	—49.0	102.86	138.1	2.634	101	29.67	14.55
A-100-175	12-30-14	100	99	92	65	0.10	—65.0	104.07	134.7	3.950	101	29.63	14.50
A-100-200	12-29-14	100	100	99	66	0.20	—47.0	105.36	130.7	4.307	100	29.75	14.60
A-108-225	12-29-14	100	108	106	74	0.20	—42.0	105.93	143.6	5.386	108	29.75	14.60
A-110-225	12-30-14	100	110	108	69	0.20	—53.0	105.43	139.7	5.457	109	29.67	14.55
A- 84-150	12-21-14	100	84	76	68	....	......	103.14	124.0	3.066	84	29.49	14.45
A-200-150	3-15-15	100	197	165	75	0.10	— 4.0	102.64	120.9	2.036	203	29.19	14.30
A-200-175	3-15-15	100	199	166	75	0.10	— 1.0	102.80	130.3	2.629	203	29.19	14.30
A-200-200	3-15-15	100	198	168	70	0.10	— 3.0	103.00	138.3	3.437	203	29.19	14.30
A-200-200	3-10-15	100	199	179	78	0.20	— 9.0	102.80	146.6	3.437	203	29.52	14.50
A-200-215	3-10-15	100	201	185	80	0.20	— 8.0	103.50	151.4	3.810	203	29.52	14.50
A-300-150	3-10-15	100	291	251	80	0.25	23.0	101.80	140.6	1.588	306	29.55	14.50
A-300-175	3-10-15	100	293	257	78	0.10	21.0	101.86	134.1	2.027	305	29.55	14.50
A-300-200	3-10-15	100	...	261	76	0.10	20.5	102.14	132.6	2.576	306	29.55	14.50
A-300-215	3-10-15	100	...	259	76	0.10	20.0	102.64	136.1	3.000	304	29.55	14.50
A-400-150	3- 9-15	100	381	316	82	0.00	6.0	101.80	117.6	1.213	409	29.67	14.50
A-400-175	3- 5-15	100	390	333	86	0.05	54.5	102.14	124.6	1.513	410	28.97	14.20
A-400-200	3- 5-15	100	391	335	84	0.05	56.0	102.29	134.9	1.926	411	28.97	14.20
A-400-220	3- 8-15	100	393	339	78	0.10	52.0	102.50	126.4	2.450	411	29.62	14.50

TABLE 2 (CONTINUED)

RESULTS OF TESTS USING AIR ALONE

Designation	Engine Data								Results				
	14	15	16	17	18	19	20	21	22	23	24	25	26
	r. p. m.	m. e. p. Head End	m. e. p. Crank End	i. h. p. Head End	i. h. p. Crank End	Total i. h. p.	b. h. p.	Mechanical Efficiency	Total Air by Formula lb. per hr.	Correction Factor	Total Air Corrected lb. per. hr.	Air per i. h. p. hr. lb.	Air per b. h. p. hr. lb.
A-100-125	105.57	32.85	34.53	7.95	7.89	15.84	13.20	0.833	805	1.000	805	50.8	61.0
A-100-150	104.95	37.67	39.37	9.06	8.95	18.01	15.74	0.874	878	0.998	876	48.6	55.6
A-100-175	104.27	46.71	49.01	11.15	11.07	22.22	18.25	0.821	1080	0.990	1069	48.0	58.5
A-100-200	103.97	49.67	52.54	11.84	11.84	23.68	20.79	0.878	1139	0.987	1124	47.5	54.0
A-108-225	103.80	54.41	56.59	12.95	12.72	25.67	23.36	0.910	1258	0.982	1235	48.1	52.8
A-110-225	104.03	55.91	55.54	13.33	12.52	25.85	23.41	0.906	1263	0.981	1244	48.1	53.1
A-84-150	103.80	39.50	42.31	9.40	9.51	18.91	15.57	0.823	958	0.995	953	50.3	61.1
A-200-150	106.00	39.07	41.10	9.50	9.43	18.93	15.90	0.840	783	1.000	783	41.3	49.2
A-200-175	105.43	45.29	45.47	10.95	10.38	21.33	18.45	0.865	882	0.998	880	41.2	47.7
A-200-200	105.17	50.90	50.44	12.27	11.48	23.75	21.03	0.885	1001	0.993	994	41.8	47.2
A-200-200	104.77	49.01	51.19	11.78	11.62	23.40	20.95	0.895	995	0.993	988	42.2	47.1
A-200-215	104.77	51.77	54.56	12.44	12.38	24.82	22.53	0.908	1045	0.991	1036	41.7	46.0
A-300-150	106.17	35.99	37.99	8.75	8.73	17.48	15.93	0.911	678	1.003	680	38.9	42.7
A-300-175	105.80	42.01	43.93	10.18	10.07	20.25	18.52	0.915	770	1.001	771	38.1	41.6
A-300-200	105.23	47.67	49.39	11.50	11.25	22.75	21.05	0.925	869	0.998	867	38.1	41.2
A-300-215	104.80	50.76	53.51	12.20	12.14	24.34	22.53	0.926	935	0.995	930	38.2	41.3
A-400-150	104.90	35.26	37.54	8.48	8.53	17.01	15.74	0.925	605	1.005	608	35.8	38.6
A-400-175	103.60	41.57	42.69	9.86	9.59	19.45	18.13	0.932	676	1.003	678	34.8	37.4
A-400-200	103.00	47.21	48.57	11.14	10.83	21.97	20.60	0.938	751	1.001	752	34.2	36.5
A-400-220	103.60	52.31	52.79	12.42	11.85	24.27	22.79	0.939	855	0.998	853	35.1	37.4

ice before making adjustments for the succeeding run. Before a test was started the steam was shut off by closing the steam valve tightly with a wrench, in order to eliminate the possibility of leakage since the introduction of a very small quantity of steam would lead to erroneous results.

Readings were secured every 5 minutes, the Venturi meter data and indicator cards being taken as nearly simultaneously as possible. The readings were as follows:

(1) Net brake load in pounds.
(2) Revolutions per minute (by the revolution counter).
(3) Pressure at throttle (kept constant) in pounds per square inch.
(4) Temperature at throttle in degrees Fahrenheit.
(5) Temperature in steam chest in degrees Fahrenheit.
(6) Temperature of engine room in degrees Fahrenheit.
(7) Temperature in exhaust pipe in degrees centigrade (converted to Fahrenheit on data sheet).
(8) Pressure in exhaust pipe in pounds per square inch.
(9) Temperature of air entering Venturi meter in degrees Fahrenheit.
(10) Pressure of air entering Venturi meter in pounds per square inch.
(11) Venturi head, by manometer, in inches of mercury.
(12) Temperature of air entering heater in degrees Fahrenheit.
(13) Temperature of air entering mixing chamber in degrees Fahrenheit.
(14) Barometer height in inches of mercury.

The results of the air tests are tabulated in Table 2, in which they are designated by the letter A, followed by the temperature at the throttle and the gage pressure at the throttle; and are shown graphically by the curves in Figs. 2, 3, and 4. It will be noted that some of the points do not fall upon the curves selected, for which fact the following explanation is offered. Variations in the initial air temperature existed, since it was impossible always to keep this temperature constant. For some tests the average temperature fell above or below the desired value, in which cases the curve was made to pass at the proper distance from the point recorded. Three such points appear on the curve which records temperature of 100 degrees Fahrenheit. The points for air at 200 degrees are somewhat scattered, hence the curve was drawn to make it consistent with the three other curves.

The curves show that with compressed air the greatest economy was secured at approximately 22 indicated horse power, which is somewhat lower than that for the greatest economy with steam as the working medium.

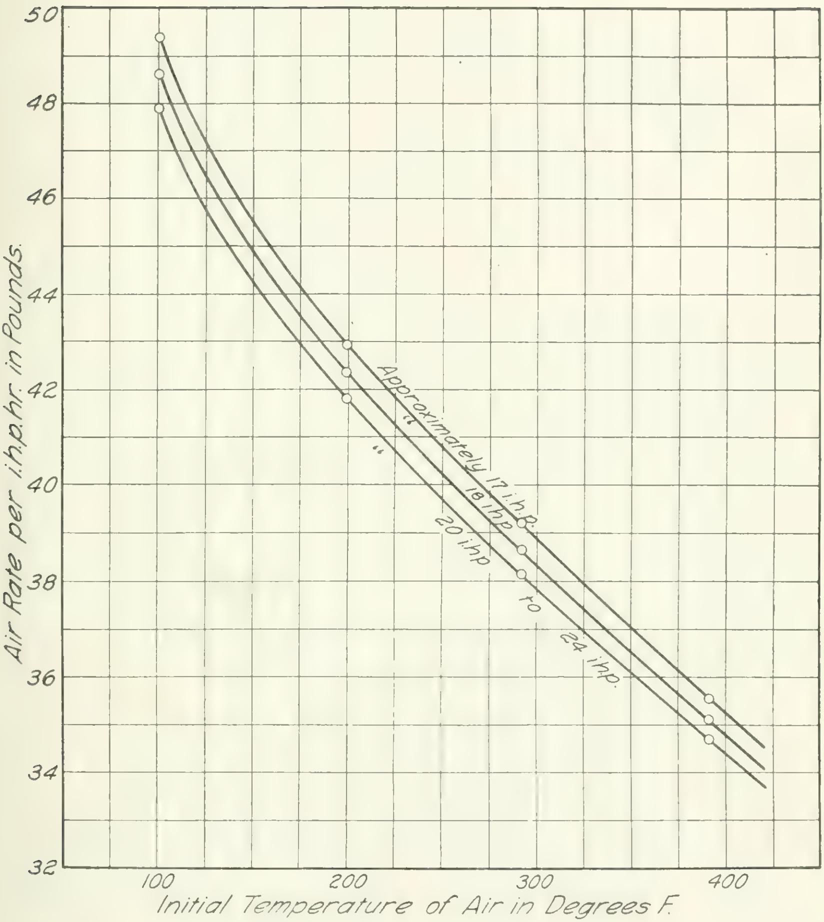

FIG. 2. AIR CONSUMPTION PER INDICATED HORSE POWER PER HOUR FOR GIVEN HORSE POWER FOR DIFFERENT INITIAL TEMPERATURES, USING AIR ALONE

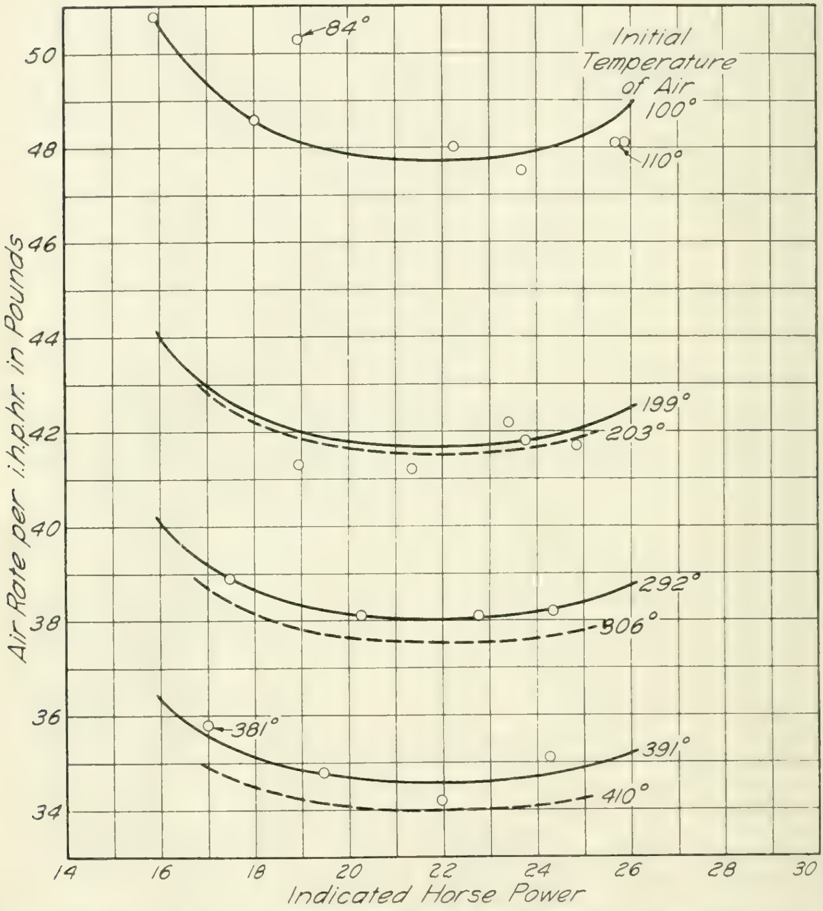

Fig. 3. Air Consumption per Indicated Horse Power per Hour for Different Initial Temperatures, Using Air Alone

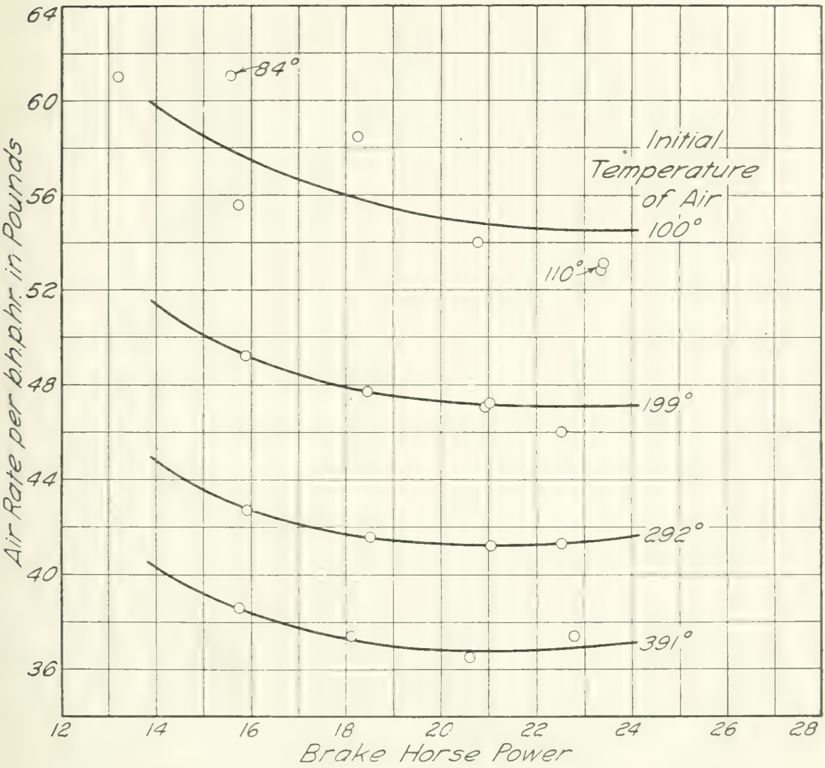

FIG. 4. AIR CONSUMPTION PER BRAKE HORSE POWER PER HOUR FOR DIFFERENT INITIAL TEMPERATURES, USING AIR ALONE

6. *Tests with Air-Steam Mixtures.*—After the usual preliminary period of adjustment of apparatus each test with mixtures of air and steam was continued for one hour. A pressure of slightly more than 100 pounds gage was maintained in the steam main, the amount of steam to the engine being controlled by an auxiliary valve located near the mixing chamber. The amount of air was regulated by throttling the air compressor. It was found that the proportion of the mixture could be judged and regulated fairly accurately from observations of the temperature just below the mixing chamber. The actual temperature of the mixture at that point proved to be practically identical with the theoretical temperature, where the initial temperatures of the air were 100 and 200 degrees, as given in Table

TABLE 3

THEORETICAL TEMPERATURE OF AIR-STEAM MIXTURES FOR DIFFERENT INITIAL AIR TEMPERATURES AND DIFFERENT PERCENTAGES OF AIR AND STEAM

Per Cent Steam in Mixture	Weight Steam in Mixture M_s lb.		Initial Air Temperatures			
			100° F.	200° F.	300° F.	400° F.
5.0	0.05263	i_a	16.46 B.t.u.	40.66 B.t.u.	64.86 B.t.u.	89.06 B.t.u.
		i_s	62.66 "	62.66 "	62.66 "	62.66 "
		i_m	79.12 "	103.32 "	127.52 "	151.72 "
		t_m	173.0° F.	208.5° F.	299.5° F.	390.0° F.
10.0	0.111111	i_a	16.46 B.t.u.	40.66 B.t.u.	64.86 B.t.u.	89.06 B.t.u.
		i_s	132.29 "	132.29 "	132.29 "	132.29 "
		i_m	148.75 "	172.95 "	197.15 "	221.35 "
		t_m	209.8° F.	219.2° F.	299.5° F.	381.3° F.
20.0	0.25	i_a	16.46 B.t.u.	40.66 B.t.u.	64.86 B.t.u.	89.06 B.t.u.
		i_s	297.65 "	297.65 "	297.65 "	297.65 "
		i_m	314.11 "	338.31 "	362.51 "	386.71 "
		t_m	248.9° F.	253.3° F.	301.0° F.	367.5° F.
30.0	0.428571	i_a	16.46 B.t.u.	40.66 B.t.u.	64.86 B.t.u.	89.06 B.t.u.
		i_s	510.26 "	510.26 "	510.26 "	510.26 "
		i_m	526.72 "	550.92 "	575.12 "	599.32 "
		t_m	272.6° F.	275.3° F.	304.5° F.	357.2° F.
40.0	0.666667	i_a	16.46 B.t.u.	40.66 B.t.u.	64.86 B.t.u.	89.06 B.t.u
		i_s	793.75 "	793.75 "	793.75 "	793.75 "
		i_m	810.20 "	834.40 "	858.60 "	882.80 "
		t_m	288.5° F.	290.0° F.	308.1° F.	350.0° F.
50.0	1.0	i_a	16.46 B.t.u.	40.66 B.t.u.	64.86 B.t.u.	89.06 B.t.u.
		i_s	1190.62 "	1190.62 "	1190.62 "	1190.62 "
		i_m	1207.10 "	1231.30 "	1255.50 "	1279.70 "
		t_m	301.4° F	302.5° F.	312.7° F.	344.3° F.
60.0	1.5	i_a	16.46 B.t.u.	40.66 B.t.u.	64.86 B.t.u.	89.06 B.t.u.
		i_s	1785.93 "	1785.93 "	1785.93 "	1785.93 "
		i_m	1802.40 "	1826.60 "	1850.80 "	1875.00 "
		t_m	311.1° F.	311.5° F.	318.0° F	341.2° F.
80.0	4.0	i_a	16.46 B.t.u.	40.66 B.t.u.	64.86 B.t.u.	89.06 B.t.u.
		i_s	4761.28 "	4761.28 "	4761.28 "	4761.28 "
		i_m	4777.70 "	4801.90 "	4826.10 "	4850.30 "
		t_m	329.4° F.	329.6° F.	329.7° F.	338.4° F.

Pressure of Mixture = 114.5 lb. abs. $t_s = 337.75°$

$i_a = 0.242(t_a - 32°)$ $i_s = i''M_s = 1190.62M_s$ $M_a = 1$ lb.

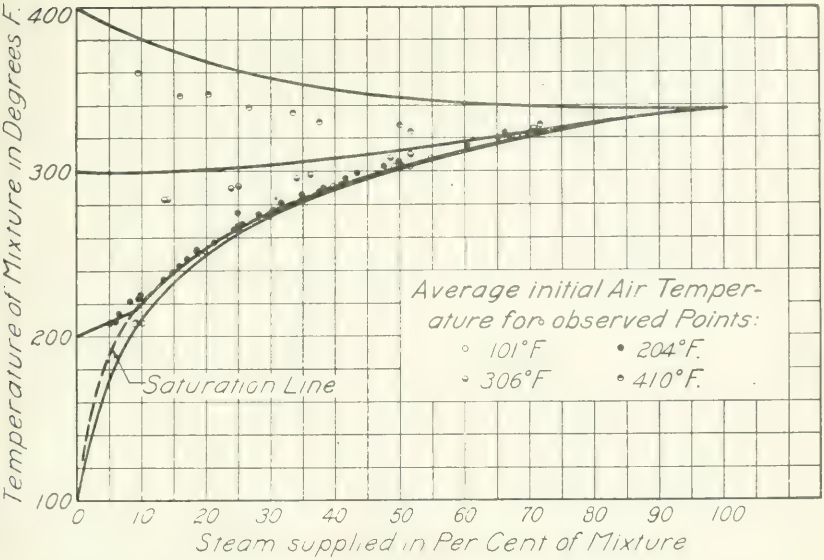

FIG. 5. TEMPERATURE OF AIR-STEAM MIXTURES FOR DIFFERENT INITIAL AIR TEMPERATURES AND DIFFERENT PERCENTAGES OF AIR AND STEAM

3 and Fig. 5. For higher initial air temperatures, however, the actual temperature of the mixture fell slightly below the theoretical, probably because of increased radiation losses at these higher temperatures.

Indicator diagrams and the following data were secured every 5 minutes, the brake load and throttle pressure being kept constant:

(1) Revolutions per minute (by revolution counter).
(2) Temperature at throttle in degrees Fahrenheit.
(3) Temperature in steam chest in degrees Fahrenheit.
(4) Temperature of engine room in degrees Fahrenheit.
(5) Temperature in exhaust pipe in degrees Fahrenheit.
(6) Temperature in calorimeter in degrees Fahrenheit.
(7) Pressure of steam entering calorimeter in pounds per square inch (steam main pressure).
(8) Temperature of air entering Venturi meter in degrees Fahrenheit.
(9) Pressure of air entering Venturi meter in pounds per square inch.
(10) Venturi head, by manometer, in inches of mercury.
(11) Temperature of air entering heater in degrees Fahrenheit.
(12) Temperature of air entering mixing chamber in degrees Fahrenheit.
(13) Temperature of condensed steam in degrees Fahrenheit.
(14) Barometer height in inches of mercury (once during test).

The weight of condensed steam was taken at the end of each run. It consisted of the water coming from the condenser, together with that carried by the saturated air exhausting from the condenser and that separated by condensation in the mixing chamber, resulting from heating the air in the tests in which air at a temperature below that of the steam was used. The first and the last of these quantities were obtained by weighing; whereas the amount of water carried off by the air was computed from its temperature and pressure.

The high-pressure air was comparatively cool at the point of entrance to the heater. It was undoubtedly saturated with water vapor. The temperature of the air being known, the partial pressure of the vapor may be determined from the steam tables. Subtracting this from the total pressure of the mixture secures the partial pressure of the air. The volume of the air may then be computed by applying Boyle's Law, and in accordance with Dalton's Law the vapor must occupy the same volume. The specific volume of the vapor, obtained from the steam tables, divided into the total volume gives the weight of moisture per pound of air. A similar process will determine the amount of moisture in the exhaust air, the temperature of which was that of the condensate and the pressure that of the atmosphere.

The data and results of the air and steam mixture tests are given in Table 4, in which they are designated by the letters AS, followed by the temperature at the throttle, the gage pressure at the throttle, and a letter added for the sake of distinction; and are presented graphically by the various curves of Figs. 6, 7, 8, and 9.

7. *Discussion of Results.*—The curves illustrating the results of the air and steam mixture tests are in accordance with the stated observations of other experimenters regarding the gain in economy by the use of air and steam together. The curves show that the maximum economy occurs with mixtures containing about 30 per cent of steam.* The maximum gain is more pronounced when cold air, rather than air previously heated, is used. This gain becomes still greater as the load decreases. See Figs. 10 and 11. The gain at the maximum load carried with air at an initial temperature of 410 degrees Fahrenheit is 5 per cent, while the gain at approximately the same load with

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* See pages 31-33.

TABLE 4

RESULTS OF TESTS USING AIR-STEAM MIXTURES

	Miscellaneous Data								Air Data					Steam Data											Engine Data							
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32
Designation	Date	Pressure at Throttle Valve lb. per sq. in. gage	Temp at Throttle Valve deg. F.	Temp in Valve Chest deg. F.	Temp. of Engine Room deg. F.	Pressure in Exhaust Pipe lb. per sq in. gage	Temp. in Exhaust Pipe deg. F.	Barometer in Hg.	Atmospheric Pressure lb. per sq in.	Pressure at Venturi Meter lb. per sq in. gage	Temp. at Venturi Meter deg. F.	Venturi Head by Manometer in. Hg.	Temp. Entering Heater deg. F.	Temp Entering Mixer deg. F.	Temp Leaving Condenser deg. F.	Weight from Condenser lb. per hr	Weight from Mixing Chamber lb. per hr.	Weight Carried off by Sat'd Exhaust Air lb. per hr	Weight Moisture Originally in Air lb. per hr	Weight Wet Steam Supplied lb. per hr	Pressure at Calorimeter lb. per sq in. gage	Temp in Calorimeter deg. F.	Quality of Steam	Heat Content per lb.	r p m	m e p Head End	m e p Crank End	i h p Head End	i h p Crank End	Total i h p	[illegible]	[illegible]
AS-10[illegible]	1-18-15	100	319	317	79.0	0.10	196.0	29.08	14.25	100.20	77.7	0.1016	67.0	97	88.0	336.2	0.4	5.4	0.3	347.7	101.0	279	0.9925	1184.0	106.65	35.08	37.04	8.57	[illegible]	17.13	16.66	[illegible]
AS-100-[illegible]	1-18-15	100	303	301	77.0	0.15	185.0	29.03	14.20	100.35	86.2	0.2325	70.0	101	85.0	253.0	7.0	7.3	0.5	266.8	101.0	279	0.9925	1184.0	106.43	35.35	35.71	8.62	8.23	16.85	15.96	[illegible]
AS-100-[illegible]	1-18-15	100	291	288	77.0	0.20	175.0	29.02	14.20	100.20	88.8	0.3530	74.5	100	81.0	206.0	7.5	7.9	0.8	220.6	102.0	278	0.9920	1183.5	106.25	35.77	36.33	8.70	[illegible]	17.06	15.94	[illegible]
AS-[illegible]	1-18-15	100	275	271	77.0	0.25	162.0	29.00	14.20	100.60	93.8	0.5225	78.0	101	78.0	157.4	7.7	8.6	1.0	172.7	101.0	279	0.9925	1184.0	106.25	35.25	37.02	8.58	8.52	17.10	15.94	[illegible]
AS-1[illegible]	1-18-15	100	251	246	76.5	0.30	143.0	28.99	14.20	101.30	103.5	0.7640	81.5	102	71.5	107.2	4.9	8.3	1.4	119.0	107.0	281	0.9925	1185.0	105.98	36.31	36.44	8.82	8.36	17.18	15.90	[illegible]
AS-1[illegible]	1-18-15	100	208	201	75.0	0.40	101.0	28.98	14.20	101.30	113.6	1.2600	89.5	100	64.0	48.8	5.1	8.1	2.2	59.8	107.0	281	0.9925	1185.0	105.82	36.79	36.50	8.95	8.37	17.32	15.57	[illegible]
AS-100-17[illegible]	1-13-15	100	321	319	81.0	0.15	197.5	29.17	14.30	100.40	84.1	0.1140	75.0	98	90.0	395.0	7.4	6.0	0.4	408.0	101.5	279	0.9925	1184.0	106.20	41.46	43.24	10.08	9.95	20.03	18.79	[illegible]
AS-100-17[illegible]	1-13-15	100	303	301	84.0	0.30	185.0	29.16	14.30	100.50	89.3	0.3020	78.0	101	85.0	283.0	7.5	8.2	0.8	297.9	102.0	279	0.9925	1184.0	106.08	40.10	42.70	9.73	9.82	19.55	18.56	[illegible]
AS-10[illegible]	1-13-15	100	291	288	85.0	0.35	175.0	29.14	14.30	100.60	93.1	0.4610	82.0	101	82.0	232.8	8.1	9.5	1.1	249.3	102.0	279	0.9925	1184.0	105.95	40.40	42.90	9.81	9.85	19.66	18.74	[illegible]
AS-1[illegible]	1-13-15	100	277	273	85.0	0.40	164.0	29.11	14.30	100.80	106.4	0.6520	87.0	102	80.5	186.2	6.9	10.3	1.5	201.9	102.0	279	0.9925	1184.0	105.88	39.86	43.09	9.67	9.87	19.54	18.53	[illegible]
AS-10[illegible]	1-13-15	100	250	244	85.0	0.50	142.0	29.10	14.30	101.40	123.4	1.0550	95.0	101	72.0	120.4	5.6	9.7	2.4	133.3	108.0	282	0.9925	1185.0	105.70	40.27	42.53	9.76	9.74	19.50	18.50	[illegible]
AS-10[illegible]	1-13-15	100	208	202	80.5	0.60	103.0	29.12	14.30	101.60	126.9	1.5850	100.0	102	62.5	57.0	4.8	8.5	3.4	76.7	107.5	282	0.9925	1185.0	105.45	40.81	42.97	9.85	9.82	19.67	18.45	[illegible]
AS-10[illegible]	1-12-15	100	321	319	83.0	0.20	198.0	29.35	14.40	100.40	92.0	0.1348	78.0	100	95.0	447.8	5.5	7.7	0.5	460.5	102.0	279	0.9925	1184.0	105.44	47.52	48.45	11.48	11.06	22.54	21.09	[illegible]
AS-1[illegible]	1-12-15	100	303	302	84.0	0.30	187.0	29.34	14.40	101.30	94.1	0.3480	80.0	101	92.5	335.0	5.7	11.4	0.9	351.2	104.0	280	0.9925	1184.5	105.23	46.20	47.63	11.15	10.85	22.00	21.05	[illegible]
AS-[illegible]	1-12-15	100	291	289	86.0	0.40	178.0	29.32	14.40	101.30	115.7	0.5730	88.0	102	89.0	271.2	6.3	12.7	1.5	288.7	107.0	281	0.9925	1185.0	105.12	46.90	48.27	11.30	11.00	22.30	21.02	[illegible]
AS-[illegible]	1-12-15	100	275	272	73.0	0.50	165.0	29.30	14.40	101.40	121.3	0.8790	95.0	101	85.0	206.7	7.1	13.8	2.2	225.4	117.0	285	0.9930	1186.5	105.02	45.96	48.26	11.06	10.98	22.04	21.00	[illegible]
AS-1[illegible]	1-12-15	100	250	246	80.0	0.60	143.5	29.31	14.40	101.90	127.0	1.3400	100.0	103	75.5	135.0	5.9	12.3	3.2	150.0	115.0	284	0.9930	1186.5	105.07	45.24	48.74	10.90	11.10	22.00	21.01	[illegible]
AS-10[illegible]	1-12-15	100	209	203	81.0	0.80	103.0	29.32	14.40	102.80	138.7	2.0500	105.0	102	65.0	64.3	6.5	10.4	4.5	76.7	115.0	284	0.9930	1186.5	104.93	45.73	48.89	11.00	11.13	22.13	20.99	[illegible]
AS-10[illegible]	1-11-15	100	323	320	78.0	0.40	199.0	29.28	14.30	100.25	85.5	0.1613	80.0	103	101.0	516.0	5.4	10.4	0.6	531.2	102.0	279	0.9925	1184.0	104.86	52.65	53.65	12.65	12.20	24.85	23.59	0.949
AS-1[illegible]	1-11-15	100	308	306	82.0	0.40	190.0	29.25	14.30	100.40	89.0	0.3745	81.0	101	97.0	398.5	7.5	13.9	1.0	418.9	102.5	279	0.9925	1184.0	104.65	51.76	52.76	12.42	11.96	24.38	23.55	[illegible]
AS-[illegible]	1-11-15	100	289	287	81.0	0.60	176.0	29.25	14.30	101.70	106.0	0.7870	85.0	100	92.0	294.3	6.5	16.7	1.5	316.0	113.0	283	0.9925	1186.0	104.68	52.76	54.09	12.66	12.27	24.93	23.55	[illegible]
AS-1[illegible]	1-11-15	100	273	270	78.0	0.80	164.0	29.25	14.30	101.60	132.3	1.2110	90.0	101	88.0	227.2	7.4	17.7	2.2	250.1	114.0	283	0.9925	1186.0	104.60	52.03	54.14	12.47	12.26	24.73	23.54	[illegible]
AS-[illegible]	1-20-15	100	324	322	78.0	0.35	202.0	29.16	14.30	100.30	76.2	0.1512	65.5	100	98.0	530.2	7.2	9.1	0.4	546.1	100.0	278	0.9920	1183.5	103.48	53.62	53.15	12.71	11.92	24.63	23.28	[illegible]
AS-1[illegible]	1-20-15	100	292	289	77.0	0.60	178.0	29.20	14.30	101.15	90.7	0.7220	72.0	102	91.0	308.9	7.0	15.8	1.0	330.7	107.0	281	0.9925	1185.0	103.15	52.99	53.07	[illegible]	11.87	[illegible]	23.21	[illegible]
AS-10[illegible]	1-20-15	100	249	245	74.0	0.80	146.0	29.25	14.30	101.90	105.0	1.7400	82.0	102	77.5	154.1	7.4	15.1	2.1	174.5	116.0	285	0.9930	1187.0	103.03	53.02	53.55	[illegible]	11.95	24.47	23.18	[illegible]
AS-1[illegible]	1-20-15	100	209	207	73.0	1.00	105.0	29.28	14.30	103.50	122.5	2.6550	97.0	102	63.0	72.7	6.7	11.2	4.1	86.5	111.0	283	0.9925	1186.0	103.05	52.39	53.48	[illegible]	11.93	[illegible]	23.19	[illegible]
AS-20[illegible]	2-19-15	100	243	240	72.0	0.35	138.0	29.64	14.50	100.70	121.7	0.7960	86.0	202	68.0	80.5	4.5	7.1	1.6	90.5	101.0	278	0.9920	1183.5	104.70	36.06	38.01	[illegible]	8.63	[illegible]	15.71	[illegible]
AS-2[illegible]	2-19-15	100	225	214	82.0	0.40	113.0	29.64	14.50	100.80	131.2	0.9950	101.0	204	64.0	50.6	3.6	6.8	2.8	58.2	102.0	279	0.9925	1184.0	104.48	35.50	37.87	[illegible]	8.58	17.08	15.68	[illegible]
AS-[illegible]	2-19-15	100	209	194	82.0	0.45	91.0	29.63	14.50	100.70	129.8	1.1900	108.0	202	59.5	31.9	2.9	6.4	3.8	37.4	101.0	279	0.9925	1184.0	104.40	36.69	39.25	[illegible]	8.88	[illegible]	15.67	[illegible]
AS-2[illegible]	2-18-15	100	303	299	79.0	0.20	185.0	29.63	14.50	99.90	103.2	0.2120	77.0	202	78.0	236.5	4.6	5.4	0.6	245.9	101.0	275	0.9910	1182.5	104.82	35.02	36.22	8.41	8.22	16.63	15.74	[illegible]
AS-2[illegible]	2-18-15	100	288	284	85.0	0.25	175.0	29.62	14.50	100.35	115.7	0.3445	85.0	203	78.0	183.4	4.5	6.7	1.0	193.6	102.0	275	0.9910	1182.5	104.73	35.32	36.50	8.48	8.28	16.76	15.72	[illegible]
AS-2[illegible]	2-18-15	100	267	262	86.0	0.40	156.0	29.60	14.50	100.20	121.3	0.5500	95.0	203	73.0	125.3	4.0	7.1	1.8	134.6	102.0	275	0.9910	1182.5	104.65	34.94	36.84	[illegible]	8.35	[illegible]	15.76	0.938
AS-2[illegible]	2-18-15	100	252	244	80.0	0.30	142.0	29.56	14.50	100.50	118.7	0.6920	97.0	203	68.5	93.5	4.4	6.9	2.1	102.7	102.0	275	0.9910	1182.5	104.55	35.18	37.06	8.42	8.40	16.82	15.69	0.933
AS-2[illegible]	2-16-15	100	323	316	83.0	0.20	198.0	29.75	14.60	100.10	96.0	0.1282	79.0	203	87.0	372.2	4.0	5.6	0.5	381.3	101.0	276	0.9910	1182.5	104.57	41.05	42.22	[illegible]	9.56	[illegible]	18.30	[illegible]
AS-20[illegible]	2-16-15	100	306	300	84.0	0.30	187.0	29.75	14.60	100.40	96.0	0.2745	80.0	203	83.5	279.5	4.3	7.3	0.8	290.3	102.0	276	0.9910	1182.5	104.45	41.15	42.09	[illegible]	9.52	19.37	[illegible]	0.943
AS-2[illegible]	2-16-15	100	296	290	84.0	0.30	180.0	29.70	14.55	100.10	103.0	0.3920	83.0	201	81.0	235.4	4.3	8.0	1.0	246.7	101.0	276	0.9910	1182.5	104.36	41.28	42.32	[illegible]	9.56	[illegible]	[illegible]	[illegible]
AS-2[illegible]	2-11-15	100	281	277	82.0	0.40	167.0	29.38	14.40	100.40	113.0	0.5940	87.0	203	78.0	182.9	4.5	9.0	1.4	195.0	101.0	279	0.9925	1184.0	104.47	41.37	42.51	[illegible]	9.62	[illegible]	[illegible]	0.936
AS-2[illegible]	2-11-15	100	274	269	81.0	0.40	162.0	29.38	14.40	100.40	106.0	0.6630	92.0	203	77.0	163.0	4.5	9.3	1.8	175.0	101.0	279	0.9925	1184.0	104.42	41.38	42.15	[illegible]	9.54	19.44	18.27	[illegible]
AS-2[illegible]	2-11-15	100	253	248	83.0	0.50	144.0	29.35	14.40	100.50	114.0	0.9390	95.0	203	71.0	109.1	4.5	8.9	2.3	120.2	101.0	279	0.9925	1184.0	104.66	41.72	42.39	10.[illegible]	9.61	19.62	[illegible]	0.933
AS-2[illegible]	2-11-15	100	235	229	82.0	0.60	128.0	29.35	14.40	100.30	118.3	1.1750	102.0	203	67.0	80.1	4.4	8.5	3.1	90.7	101.0	279	0.9925	1184.0	104.57	41.18	43.09	[illegible]	9.75	[illegible]	18.30	[illegible]
AS-2[illegible]	2-11-15	100	208	191	80.0	0.70	86.0	29.34	14.40	100.50	121.7	1.7300	104.0	203	60.0	31.[illegible]	3.0	8.0	4.1	38.3	102.0	279	0.9925	1184.0	104.33	42.22	44.54	10.[illegible]	10.06	20.[illegible]	[illegible]	[illegible]
AS-2[illegible]	2-11-15	100	221	211	78.0	0.60	109.0	29.34	14.40	100.60	122.0	1.3920	106.0	203	64.0	49.1	3.4	8.2	3.8	56.9	102.0	279	0.9925	1184.0	104.53	41.59	43.64	[illegible]	9.88	19.84	[illegible]	0.921
AS-2[illegible]	2-17-15	100	286	281	71.0	0.35	172.0	29.68	14.50	100.30	127.5	0.5300	100.0	203	79.0	199.6	5.0	8.5	2.0	211.1	101.5	277	0.9915	1183.0	104.13	41.35	42.66	[illegible]	9.62	19.49	[illegible]	[illegible]
AS-2[illegible]	2-17-15	100	268	262	73.0	0.30	158.0	29.65	14.50	100.40	132.4	0.7450	101.0	203	75.0	148.7	4.8	8.8	2.4	159.9	102.0	276	0.9910	1182.5	104.22	41.28	42.76	[illegible]	9.65	19.50	18.24	0.935
AS-2[illegible]	2-8-15	100	321	317	77.0	0.20	..	29.63	14.50	99.90	88.4	0.1162	67.5	205	85.5	376.8	4.6	5.1	0.3	386.2	101.5	279	0.9925	1184.0	105.10	39.64	42.44	[illegible]	9.68	19.23	18.39	[illegible]
AS-[illegible]	2-9-15	100	299	295	76.0	0.30	181.0	29.68	14.50	100.80	86.0	0.3400	71.0	205	82.0	240.2	4.4	7.9	0.6	251.9	101.5	279	0.9925	1184.0	103.57	39.02	43.03	[illegible]	9.65	18.91	18.12	[illegible]
[illegible]	2-15-15	100	320	316	79.0	0.30	198.0	29.48	14.45	100.35	99.3	0.1635	78.0	204	94.0	413.0	4.2	8.2	0.6	424.8	101.0	277	0.9915	1183.0	103.95	46.95	47.52	[illegible]	10.70	21.89	[illegible]	[illegible]
[illegible]	2-15-15	100	305	300	78.5	0.30	187.0	29.48	14.45	100.15	99.8	0.3535	83.0	203	88.5	314.8	4.1	9.8	1.0	327.7	102.0	274	0.9900	1182.0	103.78	46.77	47.53	[illegible]	10.68	21.80	[illegible]	[illegible]
[illegible]	2-15-15	100	290	287	80.0	0.40	176.0	29.50	14.45	100.00	104.1	0.6060	88.0	202	85.0	251.3	4.7	11.5	1.5	266.0	102.0	274	0.9900	1182.0	103.63	47.04	48.21	11.16	10.83	21.99	20.75	0.942
AS-200-200-[illegible]	2-15-15	100	265	261	81.0	0.60	156.0	29.51	14.45	100.40	113.0	1.0020	92.0	203	77.5	161.6	4.7	11.3	[illegible]	175.4	101.0	276	0.9910	1182.5	103.70	46.89	47.98	11.15	10.78	21.93	20.74	0.946
AS-2[illegible]	2-15-15	100	247	240	84.0	0.70	140.0	29.55	14.50	100.70	116.0	1.3250	95.0	203	72.5	114.8	4.3	11.0	2.8	127.3	101.0	278	0.9920	1183.5	103.68	47.36	48.46	11.25	10.89	22.14	20.74	[illegible]
A[illegible]	2-15-15	100	223	213	82.0	0.80	115.0	29.55	14.50	101.00	120.4	1.8400	100.0	203	65.0	66.2	4.4	9.9	5.7	76.8	102.0	278	0.9920	1183.5	103.43	47.82	48.78	11.34	10.93	22.27	[illegible]	[illegible]
AS-2[illegible]	2-5-15	100	303	299	77.5	0.40	183.0	29.85	14.60	100.30	103.0	0.3890	79.0	207	88.0	298.9	3.2	10.4	0.9	311.8	101.0	278	0.9920	1183.5	103.67	46.90	47.81	11.15	10.74	21.89	20.73	[illegible]
AS-200-[illegible]	2-5-15	100	289	286	76.0	0.45	173.0	29.83	14.60	100.30	106.2	0.6010	88.0	203	86.0	235.8	5.4	12.0	1.5	251.7	101.0	277	0.9915	1183.0	103.57	46.05	47.55	10.93	10.67	21.60	20.71	[illegible]
AS-200-[illegible]	2-5-15	100	275	271	75.0	0.60	162.0	29.83	14.60	100.30	108.2	0.8250	90.0	203	82.0	183.2	4.8	12.3	1.8	198.5	102.0	277	0.9915	1183.0	103.38	45.78	47.52	10.84	10.64	21.48	[illegible]	[illegible]
[illegible]	2-3-15	100	316	314	76.0	0.35	194.0	29.38	14.40	100.30	101.6	0.2200	78.0	204	89.5	387.2	4.3	8.3	0.7	399.1	101.0	278	0.9920	1183.5	104.47	45.81	48.68	10.96	11.02	21.98	20.89	[illegible]
[illegible]	2-17-15	100	277	273	78.0	0.60	166.0	29.75	14.60	100.35	118.0	0.7930	82.0	204	80.0	195.0	4.7	10.7	1.4	209.0	100.0	277	0.9915	1183.0	103.54	46.54	48.01	11.05	10.76	21.81	20.71	[illegible]
[illegible]	2-17-15	100	257	252	73.0	0.70	151.0	29.69	14.55	100.40	129.5	1.1420	97.0	203	76.0	140.6	4.8	11.3	2.7	154.0	101.5	277	0.9915	1183.0	103.45	47.14	48.61	11.17	10.90	22.07	[illegible]	[illegible]
[illegible]	2-12-15	100	...	313	81.0	0.45	196.0	29.42	14.40	100.20	98.0	0.2600	80.0	203	98.0	432.0	5.1	11.6	0.8	447.9	101.0	277	0.9915	1183.0	103.60	52.51	53.16	12.46	11.94	24.40	23.21	0.955
[illegible]	2-12-15	100	303	301	80.0	0.40	188.0	29.43	14.40	100.30	108.1	0.4455	87.0	203	96.0	353.2	4.8	14.1	1.2	370.9	101.0	277	0.9915	1183.0	103.64	52.09	52.68	12.36	11.84	24.20	23.32	[illegible]
[illegible]	2-12-15	100	292	290	82.0	0.50	179.0	29.40	14.40	100.50	107.2	0.6620	93.0	202	90.5	293.4	5.3	14.4	1.8	311.3	101.0	277	0.9915	1183.0	103.64	51.98	52.64	12.34	11.82	24.16	[illegible]	[illegible]
[illegible]-225-d	2-12-15	100	283	281	79.0	0.60	....	29.38	14.40	100.20	106.6	0.8535	94.0	203	87.0	255.8	5.0	14.6	2.1	273.3	105.0	279	0.9925	1184.0	103.70	52.05	53.06	12.37	11.93	24.30	[illegible]	[illegible]
[illegible]-225-e	2-12-15	100	265	262	80.0	0.70	160.0	29.38	14.40	101.00	111.4	1.2700	99.0	205	82.5	190.0	4.1	15.2	3.1	206.2	105.0	279	0.9925	1184.0	103.45	52.22	53.61	12.38	12.02	24.40	[illegible]	[illegible]
[illegible]-200-225-f	2-12-15	100	239	235	79.0	0.90	136.0	29.38	14.40	101.70	125.9	1.9000	107.0	203	74.0	115.0	4.5	14.0	4.7	128.8	109.0	281	0.9925	1185.0	103.37	52.70	53.61	12.48	12.00	24.48	23.26	[illegible]
[illegible]-200-225-g	2-12-15	100	214	201	77.0	1.15	101.0	29.38	14.40	102.20	133.2	2.7750	113.0	202	63.0	54.3	2.5	11.2	6.6	61.4	112.0	282	0.9925	1185.0	103.25	53.12	55.15	12.56	12.33	24.89	23.23	[illegible]
[illegible]-300-200-a	3-12-15	100	324	322	78.0	0.30	201.5	29.66	14.50	100.10	96.3	0.1202	77.0	308	91.5	460.5	2.3	6.3	0.5	468.6	102.0	279	0.9925	1184.0	106.40	46.10	49.19	11.23	11.34	22.57	[illegible]	0.942
[illegible]-300-200-b	3-12-15	100	308	300	72.0	0.30	187.0	29.66	14.50	100.30	112.2	0.3830	81.0	306	87.5	308.8	2.3	9.8	0.9	320.0	102.0	280	0.9930	1184.5	106.08	46.41	47.91	11.28	11.01	22.29	21.22	[illegible]
[illegible]-300-200-c	3-12-15	100	296	282	73.0	0.50	173.0	29.64	14.50	100.50	118.5	0.6980	90.0	305	82.5	222.1	1.8	11.1	1.7	233.3	102.0	279	0.9925	1184.0	105.98	46.69	47.92	11.34	11.00	22.34	21.20	[illegible]
[illegible]-300-200-d	3-12-15	100	290	264	78.0	0.50	158.0	29.64	14.50	100.65	120.7	0.9875	96.0	307	77.5	159.1	1.0	11.1	2.4	168.8	102.0	279	0.9925	1184.0	105.92	46.17	48.27	11.21	11.07	22.28	[illegible]	[illegible]
[illegible]-300-200-e	3-12-15	100	283	244	74.0	0.70	138.0	29.64	14.50	100.85	124.2	1.4000	102.0	305	71.5	101.9	0.0	10.7	3.5	109.1	102.0	279	0.9925	1184.0	105.90	46.81	48.87	11.35	11.21	22.56	[illegible]	[illegible]
[illegible]-300-225-a	3-11-15	100	325	324	80.0	0.40	202.0	29.57	14.50	100.60	99.8	0.1430	76.5	308	103.0	514.8	0.0	10.1	0.5	524.4	102.0	278	0.9920	1183.5	106.05	51.53	54.77	12.52	12.57	25.09	23.86	[illegible]
[illegible]-300-225-b	3-11-15	100	310	306	80.0	0.50	190.0	29.57	14.50	100.70	107.3	0.4140	81.0	305	[illegible]	365.0	0.8	13.5	1.0	378.3	102.0	279	0.9925	1184.0	105.98	50.70	53.37	12.32	12.25	24.57	23.85	[illegible]
[illegible]-300-225-c	3-11-15	100	298	286	81.0	0.60	175.0	29.57	14.50	101.00	115.0	0.7940	89.0	305	88.5	259.8	1.8	14.6	1.8	274.4	102.0	279	0.9925	1184.0	105.75	51.04	53.29	12.36	12.21	24.57	23.79	[illegible]
[illegible]-300-225-d	3-11-15	100	291	270	78.0	0.70	162.0	29.54	14.50	101.35	123.4	1.1750	98.0	306	82.0	184.0	1.0	14.[illegible]	2.8	196.3	102.0	279	0.9925	1184.0	105.40	51.28	53.68	12.38	[illegible]	[illegible]	23.72	0.963
[illegible]-300-225-e	3-15-15	100	285	243	77.0	1.00	134.0	29.67	14.50	101.80	121.4	1.8100	104.5	305	74.0	103.3	1.0	13.5	4.2	113.6	102.0	279	0.9925	1184.0	104.42	52.29	54.68	12.52	12.36	24.88	23.49	0.944
[illegible]-400-200-a	3-4-15	100	328	323	77.0	0.27	205.0	29.55	14.50	100.30	93.3	0.1100	75.5	408	96.0	462.5	0.0	7.2	0.4	469.3	102.0	275	0.9910	1182.5	103.49	46.29	47.49	10.98	10.65	21.63	20.70	[illegible]
[illegible]-400-200-b	3-4-15	100	324	303	79.0	0.30	189.0	29.50	14.45	100.30	104.2	0.3010	79.0	411	88.0	317.3	0.0	9.0	0.8	325.5	102.0	275	0.9910	1182.5	103.38	46.64	47.54	11.04	10.65	21.69	[illegible]	[illegible]
[illegible]-400-200-c	3-4-15	100	330	290	78.0	0.35	176.0	29.46	14.40	100.65	107.6	0.5240	85.0	409	83.0	231.3	0.0	10.1	1.3	240.1	102.0	277	0.9915	1183.0	103.38	47.05	47.52	11.14	10.64	21.78	[illegible]	[illegible]
[illegible]-400-200-d	3-4-15	100	339	283	79.0	0.40	163.0	29.40	14.40	100.80	115.4	0.7540	91.0	412	78.0	165.2	0.0	10.1	1.9	173.4	102.0	277	0.9915	1183.0	103.35	46.51	47.58	11.01	10.65	21.66	[illegible]	[illegible]
[illegible]-400-200-e	3-4-15	100	346	302	78.0	0.50	142.0	29.40	14.40	100.65	123.7	1.0850	100.0	406	72.5	100.2	0.0	9.9	2.8	107.3	102.0	277	0.9915	1183.0	103.20	46.81	47.46	11.06	10.60	21.66	[illegible]	[illegible]
[illegible]-400-200-f	3-4-15	100	360	318	84.0	0.60	125.0	29.87	14.60	101.10	133.2	1.3950	101.0	410	67.0	60.3	0.0	9.4	3.3	66.4	101.5	280	0.9930	1184.5	103.03	46.80	48.70	11.05	10.87	21.92	[illegible]	[illegible]
[illegible]-400-220-a	3-9-15	100	328	304	86.0	0.50	189.0	29.67	14.50	100.80	125.2	0.4150	99.0	412	96.0	338.9	0.0	13.4	1.7	350.6	102.0	280	0.9930	1184.5	104.00	50.73	52.24	12.09	11.77	23.86	[illegible]	[illegible]
[illegible]-400-220-b	3-9-15	100	336	309	82.0	0.60	172.0	29.68	14.50	101.00	125.2	0.7080	97.0	408	89.0	226.8	0.0	14.6	2.2	239.2	102.0	280	0.9930	1184.5	103.95	50.97	52.42	12.14	11.80	23.94	[illegible]	[illegible]
[illegible]-400-220-c	3-9-15	100	347	315	79.0	0.70	154.0	29.68	14.50	101.20	120.8	1.1720	85.0	410	78.5	138.5	0.0	12.6	1.9	149.2	102.0	280	0.9930	1184.5	103.88	51.10	52.56	12.16	11.82	23.98	[illegible]	[illegible]
[illegible]-400-220-d	3-8-15	100	...	320	80.0	0.80	138.0	29.62	14.50	101.30	126.5	1.4970	107.0	410	75.0	93.8	0.0	12.5	4.1	102.2	102.0	280	0.9930	1184.5	104.57	51.46	52.17	12.33	11.82	24.15	[illegible]	[illegible]
[illegible]-400-220-e	3-8-15	100		326	79.5	0.80	126.0	29.62	14.50	101.00	126.1	1.7480	107.0	410	69.5	65.0	0.0	11.2	4.4	71.8	102.0	280	0.9930	1184.5	104.10	51.69	52.25	12.32	11.77	24.09	[illegible]	[illegible]

Table 4

Results of Tests Using Air-Steam Mixtures

Steam Data										Engine Data								Results																	
15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	
Temp Leaving Condenser deg F	Weight from Condenser lb per hr	Weight from Mixing Chamber lb per hr	Weight Carried off by Sat'd Exhaust Air lb per hr	Weight Moisture Originally in Air lb per hr	Weight Wet Steam Supplied lb per hr	Pressure at Calorimeter lb per sq in gage	Temp in Calorimeter deg F.	Quality of Steam	Heat Content per lb	r p m.	m. e p. Head End	m e p Crank End	i. h. p Head End	i h p Crank End	Total i h p	b h p.	Mechanical Efficiency	Weight Air by Formula lb. per hr	Correction Factor	Weight Air Corrected lb. per hr	Weight Dry Steam Supplied lb. per hr.	Weight Mixture in Mixing Chamber lb. per hr	Dry Steam in Per Cent of Mixture in Mixing Chamber	Weight Steam Entering Cylinder lb per hr	Weight Mixture Entering Cylinder lb per hr	Steam in Per Cent of Mixture Entering Cylinder	Theor Temp. of Dry Saturation deg. F.	Air Rate at i h p of Test from Curves	Water Rate at i h p of Test from Curves	i h p Available from Air	i h p Available from Steam	i h p Available from Air and Steam Separately	i h p Gained by Mixing	Per Cent Gained by Mixing	Designation
88.0	336.2	6.4	5.4	0.3	347.7	101.0	279	0.9925	1184.0	106.65	35.08	37.04	8.57	8.56	17.13	16.00	0.934	180.5	1.006	181.6	345.0	526.6	65.5	341.6	523.2	65.3	316.3	49.20	30.56	3.69	11.29	14.98	2.15	14.35	AS-100-150-a
85.0	253.0	7.0	7.3	0.5	266.8	101.0	279	0.9925	1184.0	106.43	35.35	35.71	8.62	8.23	16.85	15.96	0.946	271.0	1.006	272.6	264.6	537.2	49.2	260.3	532.9	48.8	301.4	49.50	30.70	5.51	8.62	14.13	2.72	19.25	AS-100-150-b
81.0	206.0	7.5	7.9	0.8	220.6	102.0	278	0.9920	1183.5	106.25	35.77	36.33	8.70	8.36	17.06	15.94	0.935	333.0	1.006	335.0	218.7	553.7	39.5	213.9	548.9	38.9	289.6	49.30	30.60	6.80	7.15	13.95	3.11	22.29	AS-100-150-c
78.0	157.4	7.7	8.6	1.0	172.7	101.0	279	0.9925	1184.0	106.25	35.25	37.02	8.58	8.52	17.10	15.94	0.932	403.0	1.006	405.4	171.3	576.7	29.7	166.0	571.4	29.1	274.3	49.30	30.60	8.22	5.60	13.82	3.28	23.70	AS-100-150-d
71.5	107.2	4.9	8.3	1.4	119.0	107.0	281	0.9925	1185.0	105.98	36.31	36.44	8.82	8.36	17.18	15.90	0.925	485.0	1.006	487.9	117.9	605.8	19.5	115.5	603.4	19.1	253.0	49.20	30.55	9.91	3.86	13.77	3.41	24.75	AS-100-150-e
64.0	48.8	5.1	8.1	2.2	59.8	107.0	281	0.9925	1185.0	105.82	36.79	36.50	8.95	8.37	17.32	15.87	0.916	617.0	1.004	619.5	59.2	678.7	8.7	56.9	676.4	8.4	212.2	49.05	30.50	12.63	1.94	14.57	2.75	18.85	AS-100-150-f
90.0	395.0	7.4	6.0	0.4	408.0	101.5	279	0.9925	1184.0	106.20	41.46	43.24	10.08	9.95	20.03	18.59	0.927	190.4	1.006	191.5	404.7	596.2	67.9	401.0	592.5	68.8	318.1	47.90	29.74	4.00	13.60	17.60	2.43	13.80	AS-100-175-a
85.0	283.0	7.5	8.2	0.8	297.9	102.0	279	0.9925	1184.0	106.08	40.10	42.70	9.73	9.82	19.55	18.56	0.949	308.0	1.006	309.8	295.5	605.3	48.8	291.2	601.0	48.4	301.0	48.00	29.84	6.45	9.90	16.35	3.20	19.57	AS-100-175-b
82.0	232.8	8.1	9.5	1.1	249.3	102.0	279	0.9925	1184.0	105.95	40.40	42.90	9.81	9.85	19.66	18.54	0.943	380.0	1.006	382.3	247.2	629.5	39.2	242.3	624.6	38.8	289.5	47.95	29.84	7.97	8.28	16.25	3.41	21.00	AS-100-175-c
80.5	186.2	6.9	10.3	1.5	201.9	102.0	279	0.9925	1184.0	105.88	39.86	43.09	9.67	9.87	19.54	18.53	0.948	446.0	1.006	448.7	200.2	648.9	30.8	196.5	645.2	30.4	277.0	48.00	29.84	9.35	6.71	16.06	3.48	21.65	AS-100-175-d
72.0	120.4	5.6	9.7	2.4	133.3	108.0	282	0.9925	1185.0	105.70	40.27	42.53	9.76	9.74	19.50	18.50	0.948	561.0	1.005	563.8	132.0	695.8	19.0	130.1	693.9	18.8	252.0	48.00	29.85	11.75	4.42	16.17	3.33	20.60	AS-100-175-e
62.5	57.0	4.8	8.5	5.4	76.7	107.5	282	0.9925	1185.0	105.45	40.81	42.97	9.85	9.82	19.67	18.45	0.938	683.0	1.003	685.0	76.0	761.0	10.0	65.5	750.5	8.7	214.0	47.95	29.80	14.26	2.55	16.81	2.86	17.00	AS-100-175-f
95.0	447.8	5.5	7.7	0.5	460.5	102.0	279	0.9925	1184.0	105.44	47.52	48.45	11.48	11.06	22.54	21.09	0.935	205.0	1.006	206.2	457.0	663.2	69.0	455.5	661.7	68.8	319.0	47.75	29.43	4.32	15.53	19.85	2.69	13.55	AS-100-200-a
92.5	335.0	5.7	11.4	0.9	351.2	104.0	280	0.9925	1184.5	105.23	46.20	47.63	11.15	10.85	22.00	21.05	0.957	331.0	1.006	333.0	348.5	681.5	51.1	346.4	679.4	51.0	303.6	47.73	29.48	6.97	11.82	18.79	3.21	17.08	AS-100-200-b
89.0	271.2	6.3	12.7	1.5	288.7	107.0	281	0.9925	1185.0	105.12	46.90	48.27	11.30	11.00	22.30	21.02	0.942	416.0	1.006	418.5	286.3	704.8	40.7	283.9	702.4	40.4	291.7	47.73	29.45	8.77	9.72	18.49	3.81	20.60	AS-100-200-c
85.0	206.7	7.1	13.8	2.2	225.4	117.0	285	0.9930	1186.5	105.02	45.96	48.26	11.06	10.98	22.04	21.00	0.908	514.0	1.006	517.1	223.5	740.6	30.2	220.5	737.6	29.9	275.8	47.73	29.48	10.83	7.58	18.41	3.63	19.70	AS-100-200-d
75.5	135.0	5.9	12.3	3.2	150.0	115.0	284	0.9930	1186.5	105.07	45.24	48.74	10.90	11.10	22.00	21.01	0.955	631.0	1.004	633.5	148.6	782.1	19.0	147.3	780.8	18.8	252.4	47.73	29.48	13.26	5.04	18.30	3.70	20.20	AS-100-200-e
65.0	64.3	6.5	10.4	4.5	76.7	115.0	284	0.9930	1186.5	104.93	45.73	48.89	11.00	11.13	22.13	20.99	0.948	775.0	1.001	775.8	75.8	851.6	8.9	74.7	850.5	8.8	214.5	47.73	29.47	16.25	2.57	18.82	3.31	17.60	AS-100-200-f
101.0	516.0	5.4	10.4	0.6	531.2	102.0	279	0.9925	1184.0	104.86	52.65	53.65	12.65	12.20	24.85	23.59	0.949	226.0	1.006	227.4	527.0	754.4	69.9	526.4	753.8	69.8	319.7	48.20	29.42	4.72	17.91	22.63	2.22	9.81	AS-100-225-a
97.0	398.5	7.5	13.9	1.0	418.9	102.5	279	0.9925	1184.0	104.65	51.76	52.76	12.42	11.96	24.38	23.55	0.965	344.0	1.006	346.1	415.5	761.6	54.5	412.4	758.5	54.4	306.7	48.00	29.42	7.21	14.12	21.33	3.05	14.30	AS-100-225-b
92.0	294.3	6.5	16.7	1.5	316.0	113.0	283	0.9925	1186.0	104.68	52.76	54.09	12.66	12.27	24.93	23.55	0.945	492.0	1.006	495.0	313.3	808.3	38.8	311.0	806.0	38.5	289.3	48.20	29.42	10.26	10.65	20.91	4.02	19.25	AS-100-225-c
88.0	227.2	7.4	17.7	2.2	250.1	114.0	283	0.9925	1186.0	104.60	52.03	54.14	12.47	12.26	24.73	23.54	0.952	595.0	1.005	598.0	247.7	845.7	29.3	244.9	842.9	29.1	274.4	48.15	29.42	12.40	8.42	20.82	3.91	18.76	AS-100-225-d
98.0	530.2	7.2	9.1	0.4	546.1	100.0	278	0.9920	1183.5	103.48	53.62	53.15	12.71	11.92	24.63	23.28	0.945	220.0	1.006	221.3	542.0	763.3	71.0	539.3	760.6	70.8	320.5	48.10	29.42	4.60	18.42	23.02	1.61	6.99	AS-100-225-e
91.0	308.9	7.0	15.8	1.0	330.7	107.0	281	0.9925	1185.0	103.15	52.99	53.07	12.52	11.87	24.39	23.21	0.952	476.0	1.006	478.9	328.0	806.9	40.7	324.7	803.6	40.4	291.6	48.00	29.42	9.97	11.15	21.12	3.27	15.48	AS-100-225-f
77.5	154.1	7.4	15.1	2.1	174.5	116.0	285	0.9930	1187.0	103.03	53.02	53.55	12.55	11.95	24.47	23.18	0.946	731.0	1.002	732.5	172.9	905.4	19.1	169.2	901.7	18.8	252.2	48.05	29.42	15.22	5.88	21.10	3.37	15.95	AS-100-225-g
63.0	72.7	6.7	11.2	4.1	86.5	111.0	283	0.9925	1186.0	103.05	52.39	53.48	12.37	11.93	24.30	23.19	0.954	895.0	0.997	892.3	85.6	977.9	8.8	83.9	976.2	8.6	213.5	48.00	29.42	18.58	2.91	21.49	2.81	13.07	AS-100-225-h
68.0	80.5	4.5	7.1	1.6	90.5	101.0	278	0.9920	1183.5	104.70	36.06	38.01	8.65	8.63	17.28	15.71	0.908	486.0	1.006	488.9	89.7	578.6	15.5	87.6	576.5	15.2	241.3	42.60	30.50	11.50	2.94	14.44	2.84	19.65	AS-200-150-a
64.0	50.6	3.6	6.8	2.8	58.2	102.0	279	0.9925	1184.0	104.48	35.50	37.87	8.50	8.58	17.08	15.68	0.918	540.0	1.005	542.7	57.7	600.4	9.6	57.4	600.1	9.6	218.5	42.70	30.60	12.70	1.89	14.59	2.49	17.06	AS-200-150-b
59.5	31.9	2.9	6.4	3.8	37.4	101.0	279	0.9925	1184.0	104.40	36.69	39.25	8.78	8.88	17.66	15.67	0.887	591.0	1.005	594.0	37.0	631.0	5.9	38.3	632.3	6.0	197.0	42.35	30.38	14.02	1.22	15.24	2.44	16.00	AS-200-150-c
78.0	236.5	4.6	5.4	0.6	245.9	101.0	275	0.9910	1182.5	104.82	35.02	36.22	8.41	8.22	16.63	15.74	0.947	255.0	1.006	256.5	243.5	500.0	48.7	241.9	498.4	48.5	301.2	43.10	30.80	5.95	7.91	13.86	2.77	20.00	AS-200-150-d
78.0	183.4	4.5	6.7	1.0	193.6	102.0	275	0.9910	1182.5	104.73	35.32	36.50	8.48	8.28	16.76	15.72	0.938	322.0	1.006	323.9	191.6	515.5	37.9	190.1	514.0	37.0	287.0	43.00	30.74	7.53	6.23	13.76	3.00	21.80	AS-200-150-e
73.0	125.3	4.0	7.1	1.8	134.6	102.0	275	0.9910	1182.5	104.65	34.94	36.84	8.38	8.35	16.73	15.70	0.938	404.0	1.006	406.4	133.2	539.6	24.7	132.4	538.8	24.6	265.7	43.00	30.74	9.45	4.33	13.78	2.95	21.40	AS-200-150-f
68.5	93.5	4.4	6.9	2.1	102.7	102.0	275	0.9910	1182.5	104.55	35.18	37.06	8.42	8.40	16.82	15.69	0.933	455.0	1.006	457.7	101.6	559.3	18.2	100.4	558.1	18.3	249.8	42.95	30.70	10.65	3.31	13.96	2.86	20.50	AS-200-150-g
87.0	372.2	4.0	5.6	0.5	381.3	101.0	276	0.9910	1182.5	104.57	41.05	42.22	9.83	9.56	19.39	18.30	0.943	193.0	1.006	194.2	377.7	571.9	66.0	377.8	572.0	66.0	317.1	41.75	29.88	4.65	12.65	17.30	2.09	12.08	AS-200-175-a
83.5	279.5	4.3	7.3	0.8	290.3	102.0	276	0.9910	1182.5	104.45	41.15	42.09	9.85	9.52	19.37	18.28	0.943	292.0	1.006	293.8	287.5	581.3	49.5	386.8	680.6	56.8	302.1	41.75	29.88	7.03	9.62	16.65	2.72	16.32	AS-200-175-b
81.0	235.4	4.3	8.0	1.0	246.7	101.0	276	0.9910	1182.5	104.36	41.28	42.32	9.87	9.56	19.43	18.26	0.940	346.0	1.006	348.1	244.2	592.3	41.2	243.4	591.5	41.2	292.6	41.75	29.86	8.53	7.98	16.51	2.92	17.68	AS-200-175-c
78.0	182.9	4.5	9.0	1.4	195.0	101.0	279	0.9925	1184.0	104.47	41.37	42.51	9.91	9.62	19.53	18.28	0.936	423.0	1.006	425.5	193.3	618.8	31.2	191.9	617.4	31.1	278.0	41.70	29.84	10.20	6.48	16.68	2.85	17.08	AS-200-175-d
77.0	163.0	4.5	9.3	1.8	175.0	101.0	279	0.9925	1184.0	104.42	41.38	42.15	9.90	9.54	19.44	18.27	0.939	450.0	1.006	452.7	173.5	626.2	27.7	172.3	625.0	27.5	271.8	41.75	29.86	10.84	5.81	16.65	2.79	16.77	AS-200-175-e
71.0	109.1	4.5	8.9	2.3	120.2	101.0	279	0.9925	1184.0	104.66	41.72	42.39	10.01	9.61	19.62	18.32	0.933	531.0	1.006	534.2	119.0	653.2	18.2	118.0	652.2	18.1	250.3	41.70	29.80	12.80	3.99	16.79	2.83	16.85	AS-200-175-f
67.0	80.9	4.4	8.5	3.1	90.7	101.0	279	0.9925	1184.0	104.57	41.18	43.09	9.87	9.75	19.62	18.30	0.932	591.5	1.005	594.5	89.8	684.3	13.1	89.4	683.9	13.1	233.8	41.70	29.80	14.24	3.02	17.26	2.36	13.66	AS-200-175-g
60.0	31.4	3.0	8.0	4.1	38.3	102.0	279	0.9925	1184.0	104.33	42.22	44.54	10.09	10.06	20.15	18.26	0.905	715.0	1.002	716.5	37.8	754.3	5.0	39.4	755.9	5.2	190.1	41.60	29.70	17.23	1.27	18.50	1.65	8.92	AS-200-175-h
64.0	49.1	3.4	8.2	3.8	56.9	102.0	279	0.9925	1184.0	104.53	41.59	43.64	9.96	9.88	19.84	18.29	0.921	642.0	1.004	644.6	56.3	700.9	8.0	57.3	701.9	8.2	211.0	41.65	29.78	15.46	1.89	17.35	2.49	14.35	AS-200-175-i
79.0	199.6	5.0	8.5	2.0	211.1	101.5	277	0.9915	1183.0	104.13	41.35	42.66	9.87	9.62	19.49	18.22	0.935	394.0	1.006	396.4	209.2	605.6	34.5	208.1	604.5	34.4	283.3	41.73	29.85	9.49	7.01	16.50	2.99	18.10	AS-200-175-j
75.0	148.7	4.8	8.8	2.4	159.9	102.0	276	0.9910	1182.5	104.22	41.28	42.76	9.85	9.65	19.50	18.24	0.935	466.0	1.006	468.8	158.2	627.0	25.2	157.5	625.3	25.2	267.0	41.73	29.85	11.23	5.30	16.53	2.97	17.96	AS-200-175-k
85.5	376.8	4.6	5.1	0.3	386.2	101.5	279	0.9925	1184.0	105.10	39.64	42.44	9.55	9.68	19.23	18.39	0.956	191.0	1.006	192.2	383.5	575.7	66.5	371.9	564.1	65.9	316.9	41.80	29.90	4.60	12.83	17.43	1.80	10.33	AS-200-175-l
82.0	240.2	4.4	7.9	0.6	251.9	101.5	279	0.9925	1184.0	103.57	39.02	43.03	9.26	9.65	18.91	18.12	0.958	328.5	1.006	330.5	250.0	580.5	43.1	248.1	578.6	42.8	294.6	41.85	30.00	7.89	8.33	16.22	2.69	16.57	AS-200-175-m
94.0	413.0	4.2	8.2	0.6	424.8	101.0	277	0.9915	1183.0	103.95	46.95	47.52	11.19	10.70	21.89	20.79	0.948	225.0	1.006	226.4	421.0	647.4	65.0	421.2	647.6	65.0	316.2	41.50	29.49	5.45	14.27	19.72	2.17	11.00	AS-200-200-a
88.5	314.8	4.1	9.8	1.0	327.7	102.0	274	0.9900	1182.0	103.78	46.77	47.53	11.12	10.68	21.80	20.76	0.952	330.0	1.006	332.0	324.4	656.4	49.4	324.6	656.6	49.4	302.1	41.50	29.50	8.00	11.00	19.00	2.80	14.74	AS-200-200-b
85.0	251.3	4.7	11.5	1.5	266.0	102.0	274	0.9900	1182.0	103.63	47.04	48.21	11.16	10.83	21.99	20.73	0.942	430.0	1.006	432.6	263.0	695.6	37.8	262.8	695.4	37.8	288.2	41.50	29.48	10.42	8.92	19.34	2.65	13.70	AS-200-200-c
77.5	161.6	4.7	11.3	2.2	175.4	101.0	276	0.9910	1182.5	103.70	46.89	47.98	11.15	10.78	21.93	20.74	0.946	549.0	1.005	551.8	173.7	725.5	23.9	172.9	724.7	23.9	264.4	41.50	29.48	13.29	5.89	19.18	2.75	14.34	AS-200-200-d
72.5	114.8	4.3	11.0	2.8	127.3	101.0	278	0.9920	1183.5	103.68	47.36	48.46	11.25	10.89	22.14	20.74	0.937	630.0	1.004	632.5	126.1	758.6	16.6	125.8	758.3	16.6	245.9	41.50	29.47	15.23	4.28	19.51	2.63	13.47	AS-200-200-e
[illegible]	66.2	4.4	9.9	3.7	76.8	102.0	278	0.9920	1183.5	103.43	47.82	48.78	11.34	10.93	22.27	20.67	0.928	740.0	1.002	741.5	75.9	817.4	9.3	76.1	817.6	9.3	217.3	41.50	29.45	17.86	2.58	20.44	1.83	8.95	AS-200-200-f
88.0	298.9	3.2	10.4	1.9	311.8	101.0	278	0.9920	1183.5	103.67	46.90	47.81	11.15	10.74	21.89	20.73	0.946	345.0	1.006	347.1	309.0	656.1	47.1	309.3	656.4	47.1	299.4	41.50	29.49	8.36	10.48	18.84	3.05	16.20	AS-200-200-g
86.0	235.8	5.4	12.0	1.5	251.7	101.0	277	0.9915	1183.0	103.57	46.05	47.55	10.93	10.67	21.60	20.71	0.958	427.0	1.006	429.6	249.5	679.1	36.7	247.8	677.4	36.6	286.6	41.50	29.50	10.35	8.46	18.81	2.79	14.83	AS-200-200-h
82.0	183.2	4.8	12.3	1.8	198.5	102.0	277	0.9915	1183.0	103.38	45.78	47.52	10.84	10.64	21.48	20.68	0.962	500.0	1.006	503.0	196.6	699.6	28.1	195.5	698.5	29.0	272.7	41.50	29.52	12.10	6.66	18.76	2.72	14.49	AS-200-200-i
89.5	387.2	4.3	8.3	0.7	399.1	101.0	278	0.9920	1183.5	104.47	45.81	48.68	10.96	11.02	21.98	20.89	0.945	260.0	1.006	261.6	395.7	657.3	60.2	395.5	657.1	60.2	312.2	41.50	29.48	6.30	13.42	19.72	2.26	11.46	AS-200-200-j
[illegible]	195.0	4.7	10.7	1.4	209.0	100.0	277	0.9915	1183.0	103.54	46.54	48.01	11.05	10.76	21.81	20.71	0.948	486.0	1.006	488.9	207.0	695.9	29.7	205.7	694.6	29.6	275.6	41.50	29.50	11.77	7.01	18.78	3.03	16.13	AS-200-200-k
76.0	140.6	4.8	11.3	2.7	154.0	101.5	277	0.9915	1183.0	103.45	47.14	48.61	11.17	10.90	22.07	20.69	0.937	578.0	1.005	580.9	152.5	733.4	20.8	151.9	732.8	20.7	257.3	41.50	29.47	14.00	5.17	19.17	2.90	15.12	AS-200-200-l
[illegible]	432.0	5.1	11.6	0.8	447.9	101.0	277	0.9915	1183.0	103.60	52.51	53.16	12.46	11.94	24.40	23.31	0.955	283.0	1.006	284.7	444.0	728.7	60.9	443.6	728.3	60.8	312.9	41.70	29.33	6.82	15.14	21.96	2.44	11.10	AS-200-225-a
96.0	353.2	4.8	14.1	1.2	370.9	101.0	277	0.9915	1183.0	103.64	52.09	52.68	12.36	11.84	24.20	23.32	0.964	368.0	1.006	370.2	367.5	737.7	49.8	367.3	737.5	49.8	302.5	41.70	29.34	8.87	12.53	21.40	2.80	13.08	AS-200-225-b
90.5	293.4	5.3	14.4	1.8	311.3	101.0	277	0.9915	1183.0	103.64	51.98	52.64	12.34	11.82	24.16	23.32	0.964	449.0	1.006	451.7	308.3	760.0	40.6	307.8	759.5	40.5	291.8	41.70	29.34	10.83	10.51	21.34	2.82	13.20	AS-200-225-c
87.0	255.8	5.0	14.6	2.1	273.3	105.0	279	0.9925	1184.0	103.70	52.05	53.06	12.37	11.93	24.30	23.33	0.959	509.0	1.006	512.0	271.0	783.0	34.6	270.4	782.4	34.6	283.8	41.70	29.33	12.28	9.24	21.52	2.78	12.92	AS-200-225-d
82.5	190.0	4.1	15.2	3.1	206.2	105.0	279	0.9925	1184.0	103.45	52.22	53.61	12.38	12.02	24.40	23.28	0.954	620.0	1.004	622.5	204.3	826.8	24.7	205.2	827.7	24.8	266.4	41.70	29.33	14.90	6.97	21.87	2.53	11.55	AS-200-225-e
74.0	115.0	4.5	14.0	4.7	128.8	109.0	281	0.9925	1185.0	103.37	52.70	53.61	12.48	12.00	24.48	23.26	0.948	751.0	1.001	752.0	127.5	879.5	14.5	129.0	881.0	14.6	239.5	41.75	29.33	18.00	4.35	22.35	2.13	9.53	AS-200-225-f
63.0	54.3	2.5	11.2	6.6	61.4	112.0	282	0.9925	1185.0	103.25	53.12	55.15	12.56	12.33	24.89	23.23	0.933	901.0	0.997	898.3	60.6	958.9	6.3	65.5	963.8	6.8	202.5	41.85	29.32	21.45	2.07	23.52	1.37	5.82	AS-200-225-g
91.5	460.5	2.3	6.3	0.5	468.6	102.0	279	0.9925	1184.0	106.40	46.10	49.19	11.23	11.34	22.57	21.28	0.942	193.5	1.006	194.7	465.0	659.7	70.5	466.8	661.5	70.5	320.4	37.50	29.43	5.19	15.80	20.99	1.58	7.53	AS-300-200-a
87.5	308.8	2.3	9.8	0.9	320.0	102.0	280	0.9930	1184.5	106.08	46.41	47.91	11.28	11.01	22.29	21.22	0.952	340.0	1.006	342.0	317.5	659.5	48.2	318.6	660.6	48.2	300.9	37.50	29.45	9.12	10.78	19.90	2.39	12.00	AS-300-200-b
82.5	222.1	1.8	11.1	1.7	233.3	102.0	279	0.9925	1184.0	105.98	46.69	47.92	11.34	11.00	22.34	21.20	0.948	456.0	1.006	458.7	231.2	689.9	33.5	233.2	691.9	33.7	282.4	37.50	29.45	12.23	7.85	20.08	2.26	11.25	AS-300-200-c
77.5	159.1	1.0	11.1	2.4	168.8	102.0	279	0.9925	1184.0	105.92	46.17	48.27	11.21	11.07	22.28	21.18	0.950	542.0	1.005	544.9	167.2	712.1	23.5	170.2	715.1	23.8	264.2	37.50	29.45	14.52	5.68	20.20	2.08	10.30	AS-300-200-d
71.5	101.9	0.0	10.7	3.5	109.1	102.0	279	0.9925	1184.0	105.90	46.81	48.87	11.35	11.21	22.56	21.18	0.938	645.0	1.005	645.6	106.2	751.8	14.1	112.6	758.2	14.8	239.5	37.50	29.42	17.20	3.61	20.81	1.75	8.42	AS-300-200-e
103.0	514.8	0.0	10.1	0.5	524.4	102.0	278	0.9920	1183.5	106.05	51.53	54.77	12.52	12.57	25.09	23.86	0.950	210.5	1.006	211.8	520.0	731.8	71.0	524.9	736.7	71.2	320.9	37.85	29.32	5.59	17.73	23.32	1.77	7.59	AS-300-225-a
96.0	365.0	0.8	13.5	1.0	378.3	102.0	279	0.9925	1184.0	105.98	50.70	53.37	12.32	12.25	24.57	23.85	0.970	355.0	1.006	357.1	375.0	732.1	51.2	378.5	735.6	51.4	304.2	37.70	29.33	9.46	12.78	22.24	2.33	10.47	AS-300-225-b
88.5	259.8	1.8	14.6	1.8	274.4	102.0	279	0.9925	1184.0	105.75	51.04	53.29	12.36	12.21	24.57	23.79	0.968	489.0	1.006	491.9	272.0	763.9	35.6	274.4	766.3	35.8	285.5	37.70	29.33	13.05	9.27	22.32	2.35	10.53	AS-300-225-c
82.0	[illegible]	1.0	14.1	2.8	196.3	102.0	279	0.9925	1184.0	105.40	51.28	53.68	12.38	12.25	24.63	23.72	0.963	592.0	1.005	594.9	194.4	789.3	24.6	198.1	793.0	25.0	266.8	37.75	29.33	15.75	6.62	22.37	2.26	10.10	AS-300-225-d
74.0	103.3	1.0	13.5	4.2	113.6	102.0	279	0.9925	1184.0	104.42	52.29	54.68	12.52	12.36	24.88	23.49	0.944	736.0	1.002	737.5	112.3	849.8	13.2	116.8	854.3	13.7	236.2	37.80	29.32	19.50	3.83	23.33	1.55	6.64	AS-300-225-e
96.0	462.5	0.0	7.2	0.4	469.3	102.0	275	0.9910	1182.5	103.49	46.29	47.49	10.98	10.65	21.63	20.70	0.957	185.0	1.006	186.1	465.0	651.1	71.4	469.7	655.8	71.6	321.1	34.00	29.50	5.47	15.77	21.24	0.39	1.84	AS-400-200-a
88.0	317.3	0.0	9.0	0.8	325.5	102.0	275	0.9910	1182.5	103.38	46.64	47.54	11.04	10.65	21.69	20.68	0.953	305.0	1.006	306.8	322.5	629.3	51.2	326.3	633.1	51.5	304.3	34.00	29.50	9.02	10.93	19.95	1.74	8.70	AS-400-200-b
83.0	231.3	0.0	10.1	1.3	240.1	102.0	277	0.9915	1183.0	103.38	47.05	47.52	11.14	10.64	21.78	20.68	0.950	400.0	1.006	402.4	237.8	640.2	37.1	241.4	643.8	37.5	287.8	34.00	29.50	11.84	8.06	19.90	1.88	9.45	AS-400-200-c
78.0	165.2	0.0	10.1	1.9	173.4	102.0	277	0.9915	1183.0	103.35	46.51	47.58	11.01	10.65	21.66	20.67	0.953	477.0	1.006	479.9	171.8	651.7	26.3	175.3	655.2	26.7	270.2	34.00	29.50	14.10	5.82	19.92	1.74	8.73	AS-400-200-d
72.5	100.2	0.0	9.9	2.8	107.3	102.0	277	0.9915	1183.0	103.20	46.81	47.46	11.06	10.60	21.66	20.64	0.952	566.0	1.005	568.8	106.2	675.0	15.7	110.1	678.9	16.2	244.6	34.00	29.50	16.73	3.60	20.33	1.33	6.54	AS-400-200-e
67.0	60.3	0.0	9.4	3.3	66.4	101.5	280	0.9930	1184.5	103.03	46.80	48.70	11.05	10.87	21.92	20.61	0.940	637.0	1.004	639.5	65.7	705.2	9.3	69.7	709.2	9.8	219.8	34.00	29.48	18.80	2.23	21.03	0.89	4.23	AS-400-200-f
96.0	338.9	0.0	13.4	1.7	350.6	102.0	280	0.9930	1184.5	104.00	50.73	52.24	12.09	11.77	23.86	22.88	0.958	351.0	1.006	353.1	348.0	701.1	49.7	352.3	705.4	49.9	302.7	34.05	29.35	10.36	11.86	22.22	1.64	7.38	AS-400-225-a
89.0	226.8	0.0	14.6	2.2	239.2	102.0	280	0.9930	1184.5	103.95	50.97	52.42	12.14	11.80	23.94	22.87	0.955	477.0	1.006	479.9	237.0	716.9	33.0	241.4	721.3	33.4	282.0	34.05	29.35	14.10	8.08	22.18	1.76	7.93	AS-400-225-b
78.5	138.5	0.0	12.6	1.9	149.2	102.0	280	0.9930	1184.5	103.88	51.10	52.56	12.16	11.82	23.98	22.85	0.952	591.0	1.005	594.0	147.8	741.8	19.9	151.1	745.1	20.3	256.0	34.05	29.35	17.44	5.04	22.48	1.50	6.67	AS-400-225-c
75.0	93.8	0.0	12.5	4.1	102.2	102.0	280	0.9930	1184.5	104.57	51.40	52.17	12.33	11.82	24.15	23.01	0.952	665.0	1.003	667.0	101.2	768.2	13.2	106.3	773.3	13.7	235.9	34.10	29.35	19.55	3.45	23.00	1.15	5.00	AS-400-225-d
69.5	65.0	0.0	11.2	4.4	71.8	102.0	280	0.9930	1184.5	104.10	51.69	52.25	12.32	11.77	24.09	22.90	0.950	718.0	1.002	719.4	70.9	790.3	9.0	76.2	795.6	9.6	218.7	34.10	29.35	21.10	2.42	23.52	0.57	2.42	AS-400-225-e

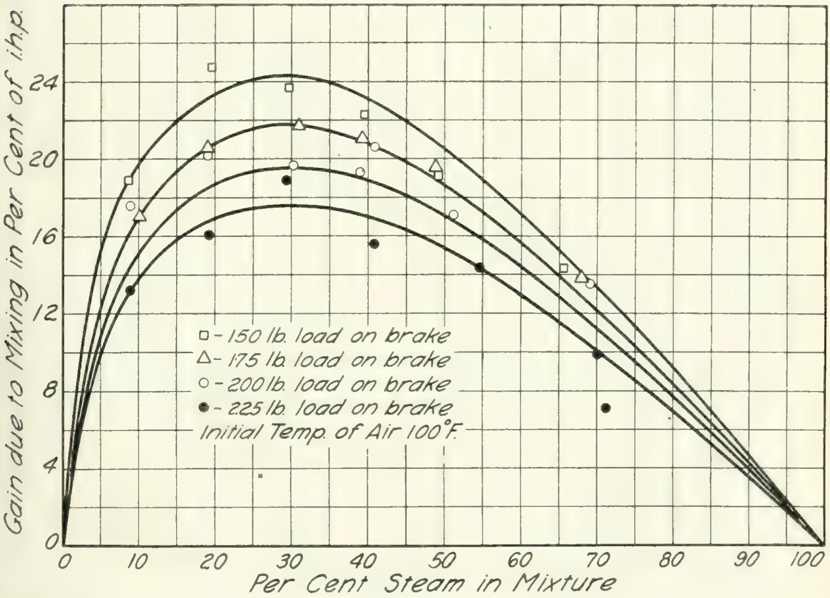

FIG. 6. PERCENTAGE GAIN IN INDICATED HORSE POWER FOR DIFFERENT PERCENTAGES OF STEAM IN MIXTURE; INITIAL AIR TEMPERATURE 100 DEG. F.

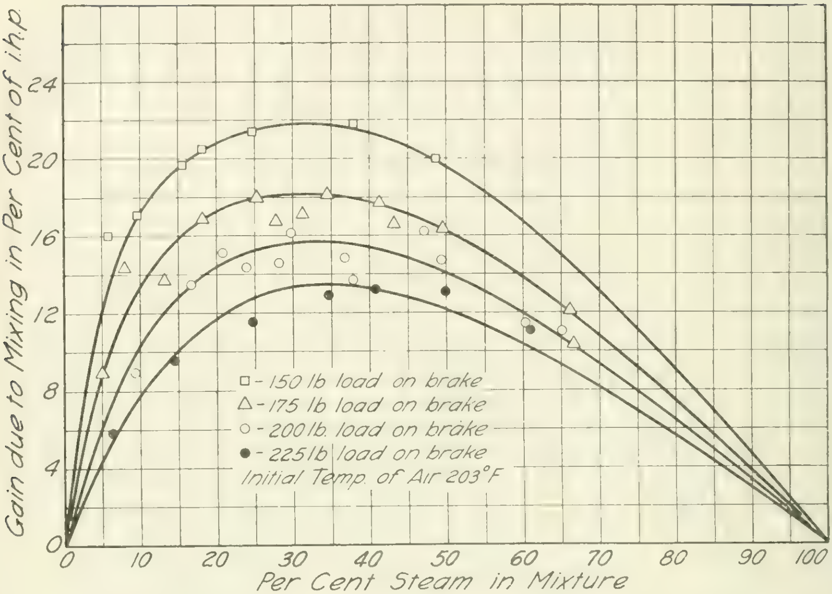

Fig. 7. Percentage Gain in Indicated Horse Power for Different Percentages of Steam in Mixture; Initial Air Temperature 203 deg. F.

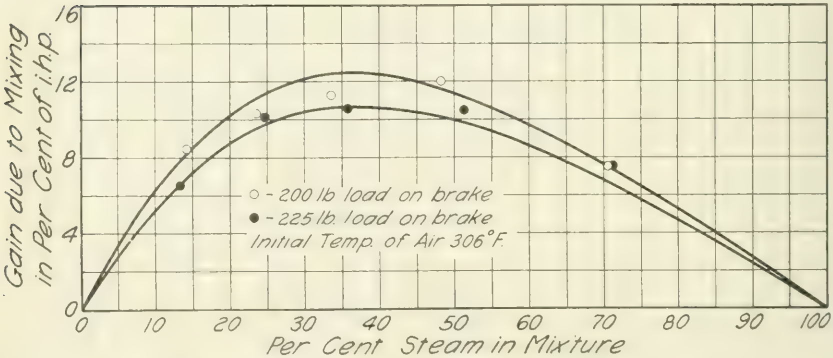

Fig. 8. Percentage Gain in Indicated Horse Power for Different Percentages of Steam in Mixture; Initial Air Temperature 306 deg. F.

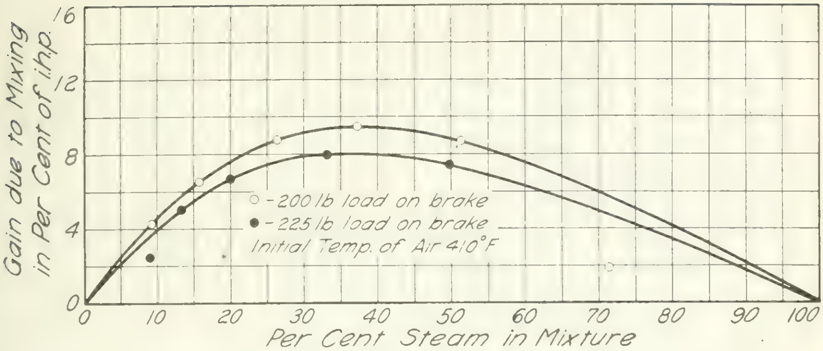

FIG. 9. PERCENTAGE GAIN IN INDICATED HORSE POWER FOR DIFFERENT PERCENTAGES OF STEAM IN MIXTURE; INITIAL AIR TEMPERATURE 410 DEG. F.

cold air at a temperature of 100 degrees Fahrenheit is about 15 per cent. At three-quarters of the maximum load and with air supplied at a temperature of 100 degrees Fahrenheit the gain is about 24 per cent. Tests at less than three-quarters of the maximum load carried were impractical because the pressures after expansion were below the back pressures on the engine, as indicated by a loop in the toe of the indicator diagram.

The results of Reynolds' experiments* furnish a partial explanation of this phenomenal gain from using a mixture of air and steam expansively. In steam engine operation, initial cylinder condensation is chiefly responsible for the great difference between the actual and the theoretical steam consumption. Any method which reduces the loss from such condensation will therefore improve the efficiency of operation. Since the total pressure of a mixture of air and steam is the sum of the partial pressures of the two constituents, the steam may be in a superheated condition in such a mixture. When such a condition exists, the initial cylinder condensation is reduced, with a consequent increase in efficiency. A further explanation of the improvement in efficiency may be found in Rankine's statement† that the air has a tendency to increase the working temperature limits of the steam, independently of the addition of heat. It may be noted, furthermore, that the presence of air reduces the effectiveness of the

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* See page 81.

† See page 80.

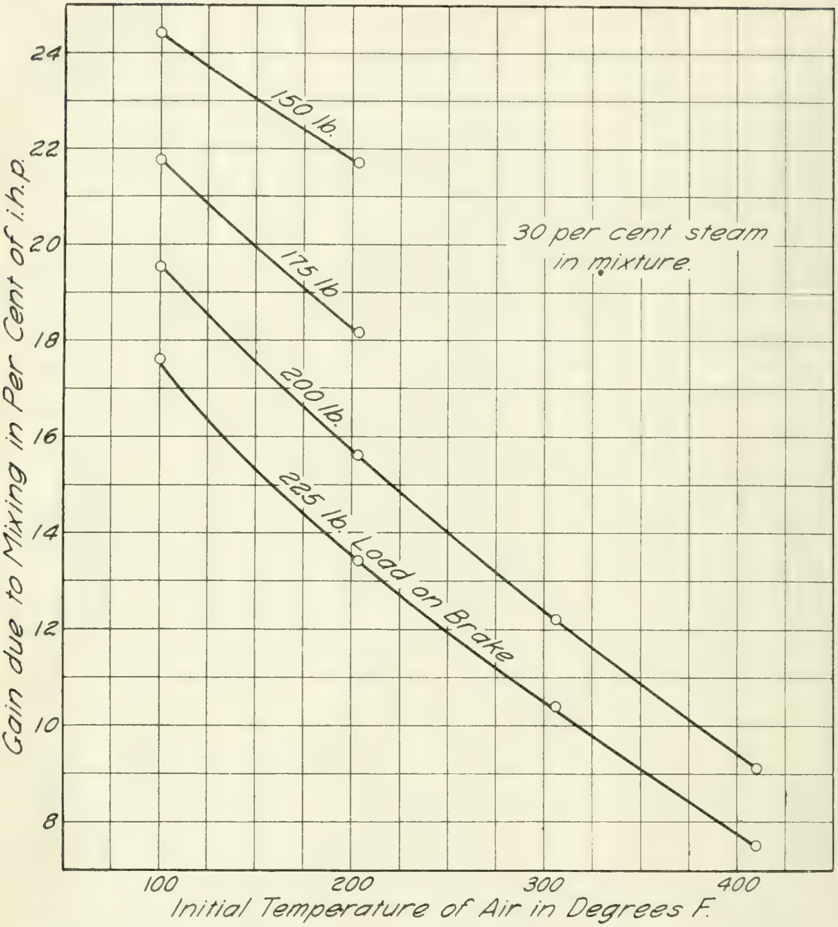

FIG. 10. PERCENTAGE GAIN IN INDICATED HORSE POWER WITH 30 PER CENT STEAM IN MIXTURE FOR DIFFERENT INITIAL AIR TEMPERATURES

interchange of heat between the working fluid and the metal walls of the cylinder, thus causing a reduction in the heat loss.

The increased gain with cold air is due in part to the preheating effect produced by the steam and to the resulting improvement in mechanical efficiency. The injection of a very small percentage of steam into the compressed air is sufficient to prevent the exhaust temperatures falling below the freezing point, as is shown by the data in Table 4.

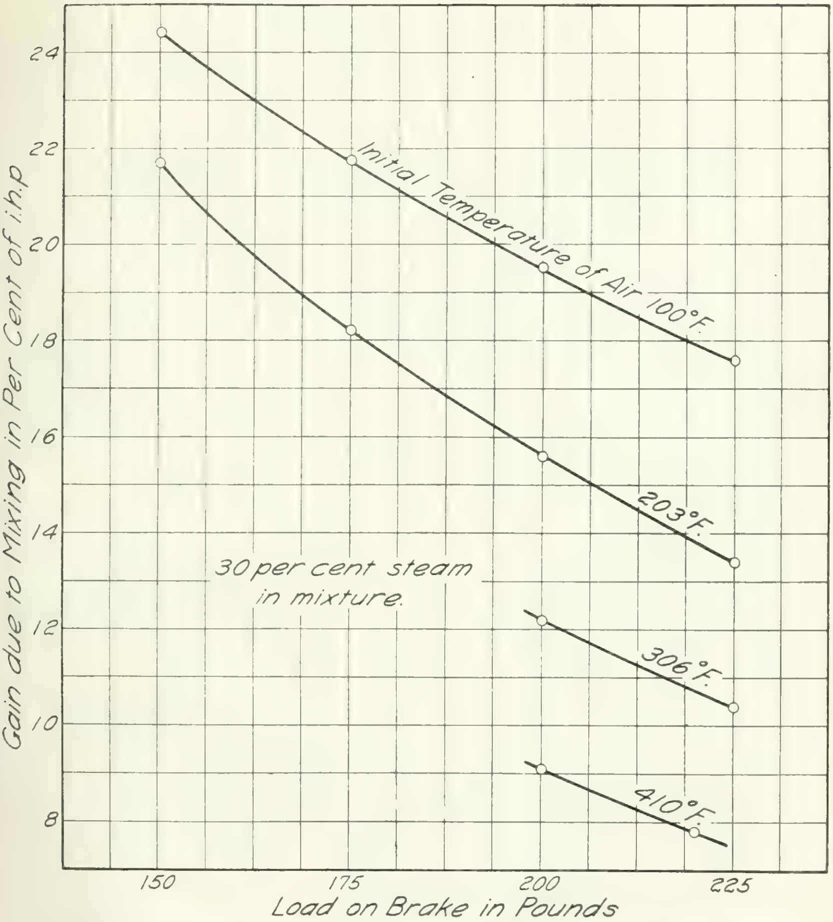

FIG. 11. PERCENTAGE GAIN IN INDICATED HORSE POWER WITH 30 PER CENT STEAM IN MIXTURE FOR DIFFERENT LOADS

III. Thermodynamics; Thermal Properties of Air-Steam Mixtures

8. *Notation.*—In general, the symbols used in this bulletin are identical with those used by G. A. Goodenough in his book "Principles of Thermodynamics."

In case a magnitude is dependent upon the weight of the substance, the small letter denotes the magnitude referred to unit weight, the capital letter the same magnitude referred to M units of weight; thus q denotes the heat absorbed by one pound of a substance, $Q = M q$, the heat absorbed by M pounds.

A subscript makes any magnitude symbolical of that particular constituent of a composite substance that the subscript represents; thus, a represents air, s steam, and w water. Lack of any subscript generally indicates that the system as a whole is under consideration. M denotes, for example, the weight of a mixture composed of M_a pounds of air, M_s pounds of dry steam, and M_w pounds of water as moisture.

Subscripts 1 and 2 in connection with any magnitude undergoing a thermodynamic change refer to the initial and the final conditions, respectively; thus U_1 denotes the energy of a system at the start, U_2 that at the finish, and $U_2 - U_1$ is the change of energy involved.

In dealing with wet steam, the thermodynamic properties of the liquid and of the vapor are distinguished by affixing a prime to designate the liquid state and a double prime for the saturated vapor state; thus, q' represents the heat of the liquid and q'' that of saturated vapor.

A = reciprocal of Joule's equivalent.
B = constant in the gas equation $pv = BT$.
c_p = specific heat of air at constant pressure.
c_v = specific heat of air at constant volume.
i, I = heat content at constant pressure.
J = Joule's equivalent.
M = weight of system under consideration.
p = partial pressure of substance.
P = total pressure of system.
q, Q = heat absorbed from external sources.

r = latent heat of vaporization.
ρ = internal latent heat.
s, S = entropy.
t = temperature on the Fahrenheit scale.
T = absolute temperature.
u, U = intrinsic energy.
v, V = volume.
W = external work.
x = quality of steam.

Other symbols used are incidental constants, the values of which are given with the equations in which the constants appear.

9. *Properties of a Mixture of Air and Steam.*—The properties of a mixture of two or more gases have been carefully studied and are well known. For comprehensive discussions of gas mixture, reference may be made to Planck's Treatise on Thermodynamics, Chap. IV, and to Partington's Text Book of Thermodynamics, Chap. X. The principal laws that apply to ideal gas mixtures are the following:

(1) Each gas occupies the whole volume, V, of the mixture.

(2) The total pressure, P, of the mixture is the sum of the partial pressures p_1, p_2, etc. of the individual gases (Dalton's law).

(3) The intrinsic energy, U, of the mixture is the sum of the intrinsic energies U_1, U_2, etc. of the individual constituents.

(4) The heat content, I, of the mixture is the sum of the heat contents I_1, I_2, etc. of the constituents.

(5) The entropy, S, of the mixture is the sum of the entropies S_1, S_2, etc. of the individual gases.

Saturated or moderately superheated steam is by no means a perfect gas; consequently laws 2 and 4 are not strictly exact for a mixture of steam and air. The deviation, however, is small, and for all practical purposes such a mixture may be treated as a mixture of ideal gases.

10. *Conditions of Mixing.*—The first fundamental problem that must be attacked in the investigation of air-steam mixtures is the following. Let M_a pounds of air at a temperature T_a and pressure P_a be mixed with M_s pounds of steam at pressure P_s, having a quality x_s, or if superheated, a temperature T_s. Required, the temperature and total pressure of the resulting mixture, and the partial pressure p_a and p_s, of the air and steam, respectively.

The solution of this problem depends upon the conditions under which the mixing occurs. Two cases will be considered:

(a) The air initially occupies the volume V_a, Fig. 12, the steam the volume V_s. The partition is removed and the steam and air are allowed to mix. It is assumed that the process is adiabatic, that is, no heat enters or leaves through the walls.

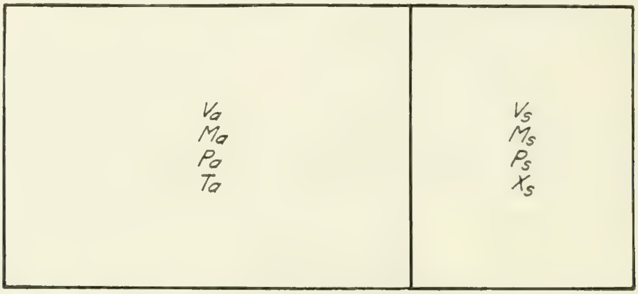

Fig. 12. Method of Forming Air-Steam Mixtures; Case A

The air expands from the volume V_a to the whole volume $V = V_a + V_s$ and the pressure drops from P_a to p_a; likewise, the steam expands from volume V_s to volume V, and the pressure drops from P_s to p_s. The sum $p_a + p_s$ gives the final pressure P of the mixture.

From the characteristic equation of gases the volume V_a is given by the equation

$$V_a = \frac{M_a B T_a}{P_a} \quad \text{. . . . .} \quad (1)$$

The volume V_s is given by the expression

$$V_s = M_s\left[x_s\,(v - v'_s) + v'_s\right] \quad \text{. . . . .} \quad (2)$$

or with sufficient accuracy by the simpler relation

$$V_s = M_s\, x_s\, v''_s \quad \text{. . . . .} \quad (3)$$

In these equations v''_s and v'_s denote respectively the specific volumes of steam and water corresponding to the pressure P_s. The total volume V is given therefore by the equation

$$V = \frac{M_a B T_a}{P_a} + M_s\, x_s\, v''_s \quad \text{. . . . .} \quad (4)$$

This volume V is occupied by the air at the partial pressure p_a and at the final temperature T of the mixture; hence the relation

$$p_a V = M_a \, B \, T \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The volume V is also occupied by the steam at the partial pressure p_s. If the final quality of the steam is denoted by x,

$$V = M_s \, x \, v'' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

in which v'' denotes the volume of 1 pound of saturated steam at the pressure p_s; if the steam in the mixture is superheated the relation becomes

$$V = M_s \, v \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

in which v denotes the volume of 1 pound of superheated steam at the pressure p_s and the final temperature T.

Since the process is adiabatic and no external work is done there is no change of energy during the process. The initial energy of the air is

$$U_a = M_a \, c_v \, T_a + U_{oa} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

and that of the steam is

$$U_s = M_s \, (u'_s + x_s \, \rho_s) + U_{os} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The unknown constants U_{oa} and U_{os} may be disregarded since they appear in each member of the final equation and cancel. The energy of the mixture is therefore

$$U = M_a \, c_v \, T_a + M_s \, (u'_s + x_s \, \rho_s) \quad . \quad . \quad . \quad . \quad . \quad (10)$$

In the mixture the energy of the air is $M_a \, c_v \, T$ and that of the steam is $M_s \, (u' + x \, \rho)$ if the quality is x, or $M_s \, u$ if the steam is superheated, and u is the energy per pound at pressure p_s and temperature T. Hence

$$U = M_a \, c_v \, T + M_s \, (u' + x\rho) \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

or

$$U = M_a \, c_v \, T + M_s \, u, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

according as the steam in the mixture is wet or superheated.

If the steam in the final mixture is wet there are three unknown quantities to be determined, namely, p_a, T, and x. These may be found from the three independent equations (5), (6), and (11). If the steam in the mixture is superheated the three unknown quantities are p_a, p_s, and T, and the three equations (5), (7), and (12) are sufficient for their determination.

(b) The air and the steam enter a chamber through separate channels and the mixture is drawn out through a third pipe, Fig. 13.

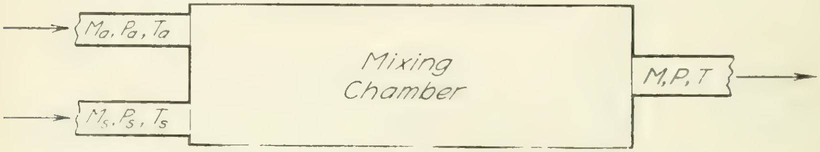

FIG. 13. METHOD OF FORMING AIR-STEAM MIXTURES; CASE B

Let M_a and M_s denote the weights of air and steam, respectively, entering the chamber per second; then with steady flow $M = M_a + M_s$ is the weight of mixture leaving per second.

In this case the relation $V_a + V_s = V$ does not hold, and there are in consequence only two instead of three equations from which to determine the unknown quantities. The fact that the volume V of M pounds of mixture is occupied by both air and steam gives the equations (5) and (6) or (5) and (7) as before, but as V is now unknown the combination of these equations gives the single relation

$$\frac{M_a B T}{p_a} = M_s x v'' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

or

$$\frac{M_a B T}{p_a} = M_s v \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

according as the steam is wet or superheated.

The second equation of condition is obtained as follows. If V_a is the volume of air entering the mixing chamber per second, the product $p_a V_a$ gives the work done on M_a pounds of air in forcing it into the chamber; and similarly the product $p_s V_s$ is the work done in M_s pounds of steam in forcing it into the chamber. In leaving the chamber the volume V flowing per second has to pass out against the external pressure P and hence does the work PV. As no heat enters or leaves, the net work done *on* the system must appear as increased energy. Taking the energy in thermal units, this fact is expressed by the equation

$$A(p_a V_a + p_s V_s - PV) = U - (U_a + U_s)$$

But since $U + A\,p\,V = I$, the equation becomes

$$I_a + I_s = I \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

That is, the heat content of the mixture is the sum of the heat contents of the air and steam, respectively, before mixing. Equations (10), (11), and (12) are in this case replaced by the following;

$$I = M_a\,c_p\,T_a + M_s\,(i'_s + x_s\,r_s) \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

$$I = M_a\,c_p\,T \;\; + M_s\,(i' + x\,r) \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

$$I = M_a\,c_p\,T \;\; + M_s i \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

As before, equation (17) is used when the steam in the mixture is wet, equation (18) when it is superheated. In (18) i denotes the heat content of one pound of superheated steam at pressure p_s and temperature T.

There being two equations of condition, only two independent unknown quantities can be determined. It is evident, indeed, that the final pressure P should not be uniquely fixed by the initial states; for its magnitude must be influenced by the rate of flow. If, therefore, P is given or assumed, the remaining unknown magnitudes are the temperature T and quality x if the steam in the mixture is wet. From the temperature the pressure p_s is known, and then $p_a = P - p_s$. If the steam in the mixture is superheated, the unknown magnitudes are T and either p_a or p_s.

11. *Illustrative Examples.*—Let 1 pound of steam at a pressure of 125 pounds per square inch and quality 0.98 be mixed with 5 pounds of air having a pressure of 125 pounds per square inch and a temperature of 100 degrees Fahrenheit. The pressure of the mixture leaving the mixing chamber is 123 pounds per square inch.

The condition that the steam and air in the mixture both occupy the total volume gives by a combination of equations (5) and (6) the equation

$$\frac{M_a\,B\,T}{p_a} = M_s\,x\,v'',$$

or

$$x = \frac{M_a}{M_s}\,\frac{B\,T}{p_a\,v''}$$

Taking p_a in pounds per square inch, and inserting the numerical values,

$$x = \frac{5 \times 53.34}{144} \frac{T}{p_a\, v''} = 1.852 \frac{T}{p_a\, v''}$$

The initial heat content of the steam (Goodenough's steam tables) is 1174.5 B.t.u.; the heat content of the air is

$$M_a\, c_p\, T_a = 5 \times 0.242 \times (100 + 460) = 677.6 \text{ B.t.u.}$$

Hence the heat content of the mixture is 1174.5 + 677.6 = 1852.1 B.t.u. If the temperature of the mixture is T, the equation for the heat content of the mixture is

$$5 \times 0.242 \times T + i' + xr = 1852.1;$$

or inserting the expression for x

$$1.21\, T + i' + \frac{1.852 T}{p_a} \frac{r}{v''} = 1852.1$$

To solve this equation assume probable values of the temperature t, and from the steam table get the corresponding values of p_s, i', r, and v''. Insert these in the equation, taking $p_a = 123 - p_s$. A preliminary trial shows that the temperature is in the vicinity of 240 degrees. Hence take 240, 245, and 250 as trial temperatures and arrange the work as follows:

	$t =$	240.0	245.0	250.0
	$T =$	700.0	705.0	710.0
From steam table	$r =$	953.9	950.6	947.3
	$v'' =$	16.33	15.02	13.83
	$i' =$	208.2	213.2	218.3
	$p_s =$	24.97	27.31	29.83
	$p_a = 123 - p_s =$	98.03	95.69	93.17
	$\frac{1.852T}{p_a} \frac{r}{v''} =$	772.5	863.6	966.7
	$i' =$	208.2	213.2	218.3
	$1.21T =$	847.0	853.1	859.1
	I of mixture $=$	1827.7	1929.9	2044.1

These values of I are plotted and the curve intersects the line $I = 1852.1$ at the temperature $t = 241.5$ nearly. This, therefore, is the temperature of the mixture.

For $t = 241.5$, the steam tables gives $p_s = 25.66$, $v'' = 15.92$; hence

$$x = \frac{1.852\, T}{p_a\, v''} = \frac{1.852 \times 701.5}{97.34 \times 15.92} = 0.838$$

As a second example, take the same data as in the preceding except that the air has a temperature of 400 degrees initially.

In this case the steam in the mixture will be superheated. The initial heat content of the air is $1.21 \times 860 = 1040.6$ B.t.u. and adding the initial heat content of the steam, the heat content of the mixture is $1040.6 + 1174.5 = 2215.1$ B.t.u. Let v denote the volume and i the heat content of 1 pound of superheated steam at the pressure p_s and temperature of the steam in the mixture. Then the two equations of condition are

$$v = \frac{1.852T}{p_a}$$

$$1.21\,T + i = 2215.1$$

The temperature of the mixture must evidently lie between 344.4 degrees, the temperature of the steam, and 400 degrees, the temperature of the air. Hence for a preliminary survey, take $t = 360$, 365, 370. The computation is arranged as in the following schedule.

$t =$	360	365	370	
$T =$	820	825	830	
$1.21T =$	992.2	998.2	1004.3	
$2215.1 - 1.21T = i =$	1222.9	1216.9	1210.8	
p_s (from table) $=$	9.5	59	105	
v (from table) $=$	51.35	8.13	4.50	(a)
$p_a = 123 - p_s =$	113.5	64	18	
$\frac{1.852T}{p_a} = v =$	13.38	23.87	85.4	(b)

With temperatures as abscissas, curves are drawn with the volumes given by rows (a) and (b) as ordinates. The intersection of these curves shows that the two values of v are coincident for $t = 362$ degrees approximately. To get a more exact value the computation is repeated, taking $t = 361$, 362, 363. The following results are then obtained: $t = 361.8$ degrees, $p_s = 28.5$ pounds, and $p_a = 94.5$ pounds per square inch.

12. *Adiabatic Changes of Air-Steam Mixtures.*—If a mixture of air and steam is subjected to an adiabatic change of state, expansion or compression, there are two problems to be considered; first, to find the condition of the mixture at the completion of the process, and second, to find the external work done by or on the mixture.

According to Law 5, Art. 9, the entropy of the mixture is the sum of the entropies of the constituents, *as they exist in the mixture.* It is *not* the sum of the entropies of the air and steam before mixing, because the mixing process always entails an increase of entropy.

Neglecting the constant S_0 that always occurs in expressions for entropy, because it disappears by subtraction, the entropy of M_a pounds of air is given by either of the three expressions

$$S_a = M_a\,(c_p \log_e T - AB \log_e P_a) \quad . \quad . \quad . \quad . \quad . \quad (19)$$

$$S_a = M_a\,(c_v \log_e T + AB \log_e v) \quad . \quad . \quad . \quad . \quad . \quad (20)$$

$$S_a = M_a\,(c_p \log_e v + c_v \log_e p_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Of these, the first expression is usually most convenient for the investigation in hand. The entropy of the steam in the mixture, if saturated or wet, is given by the usual expression

$$S_s = M_s\,(s' + \frac{xr}{T}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Hence a general expression for the entropy of the mixture is

$$S = M_a\,(c_p \log_e T - AB \log_e p_a) + M_s\left(s' + \frac{xr}{T}\right) \quad . \quad (23)$$

If the steam in the mixture is superheated its entropy is

$$S_s = M_s\,s, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

where s denotes the entropy of one pound at the given pressure and temperature. Then the entropy of the mixture is

$$S = M_a\,(c_p \log_e T - AB \log_e p_a) + M_s s \quad . \quad . \quad . \quad . \quad (25)$$

The mixture expands adiabatically from the initial pressure P_1 to a final pressure P_2. The condition of the mixture in the final state as determined by the temperature T_2, the partial pressures p_{2a} and p_{2s}, and the final quality x_2, is required. If during expansion the steam remains wet or becomes wet there will be only two unknown quantities to be determined, namely p_{2s} and x_2; for p_{2s} fixes T_2 and also p_{2a}, since $p_{2a} = p_2 - p_{2s}$. Two equations of conditions are furnished: (1) the final entropy S_2 is equal to the initial entropy S_1 given by equation (23) or equation (25). Hence

$$S_2 = S_1 = M_a\,(c_p \log_e T_2 - AB \log_e p_{a2}) + M_s\left(s_2' + \frac{x_2\,r_2}{T_2}\right) \quad (26)$$

(2) the final volume V_2 is occupied by the air and also by the steam; hence

$$V_2 = \frac{M_a\, BT_2}{p_{a2}} = M_s\, x_2\, v''_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

Eliminating x_2 between these equations, the resulting equation is

$$S_1 = M_a\,(c_p \log_e T_2 - AB \log_e p_{a2}) + M_s \left[s'_2 + \frac{M_a B\, r_2}{M_s p_{a2} v''_2} \right] \quad (28)$$

This equation contains only one unknown quantity, say p_{a2}. To determine this unknown take a series of probable values of p_{a2}, and thus get at once the corresponding values of $p_{s2} = P_2 - p_{a2}$. For these values of p_{s2} corresponding values of T_2, s'_2, r_2, and v''_2 are found in the steam tables; hence for each assumed value of p_{a2} the value for the second member of equation (28) may be calculated. These values are then plotted and the intersection of the curve with the line $S_1 =$ constant gives the desired value of p_{a2}.

If the steam in the final state is superheated the two equations of condition are:

$$S_2 = S_1 = M_a(c_p \log_e T_2 - AB\ \log_e p_{a2}) + M_s s_2, \quad . \quad (29)$$

$$\frac{M_a BT_2}{p_{a2}} = M_s v_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

in which s_2 and v_2 denote respectively the entropy and volume of one pound of superheated steam at the pressure p_{s2} and temperature T_2.

Again assume a probable value of p_{a2}; then $p_{s2} = P_2 - p_{a2}$. From equation (30)

$$v_2 = k\ T_2, \text{ where } k = \frac{M_a\ B}{M_s\ p_{a2}} \quad . \quad . \quad . \quad . \quad . \quad (31)$$

From the table of the properties of superheated steam take the pressure p_{s2} and from the corresponding values of T_2 and v_2 find the value of T_2 which satisfies the relation $v_2 = kT_2$; also find the corresponding value of s_2. The second member of equation (29) may now be calculated. Values thus obtained for different assumed values of p_{a2} are plotted, as before.

The external work performed in an adiabatic change is equal to the decrease of intrinsic energy. Thus

$$AW = U_1 - U_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

The values of U_1 and U_2 are obtained from equation (11) or equation (12).

13. *Isothermal Changes of State.*—If the steam in the mixture remains wet during the change the following relations are readily obtained. Since T remains constant, the change of energy from equation (11) is

$$U_2 - U_1 = M_s\rho\,(x_2 - x_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

From equation (23) the change of entropy is

$$S_2 - S_1 = M_a AB \log_e \frac{p_{a1}}{p_{a2}} + M_s \frac{r}{T}(x_2 - x_1) \quad . \quad . \quad (34)$$

The heat added is

$$Q = T(S_2 - S_1)$$

$$= M_a ABT \log_e \frac{p_{a1}}{p_{a2}} + M_s r\,(x_2 - x_1) \quad . \quad . \quad (35)$$

Subtracting equation (33) from equation (35), the difference gives the heat equivalent of the external work. Thus

$$AW = M_a ABT \log_e \frac{p_{a1}}{p_{a2}} + M_s (r - \rho)(x_2 - x_1) \quad . \quad . \quad (36)$$

or $$W = p_{a1}V_1 \log_e \frac{p_{a1}}{p_{a2}} + p_s (V_2 - V_1) \quad . \quad . \quad . \quad (37)$$

Since T remains constant the pressure p_s of the steam remains constant; hence if the final pressure is P_2, $p_{a2} = P_2 - p_s$. The initial and final volumes are, respectively,

$$V_1 = M_s x_1 v'', \quad V_2 = M_s x_2 v''$$

whence $$x_2 = \frac{V_2}{V_1} x_1$$

or $$x_2 = \frac{p_{a1}}{p_{a2}} x_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

If the steam in the second state is superheated, the constant temperature T is known, as is also the final pressure P_2. However, the partial pressures p_{a2} and p_{s2} are unknown. The relation

$$V_2 = M_s v_2 = \frac{M_a BT}{p_{a2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

gives again $v_2 = k T$, where $k = \frac{M_a B}{M_s p_{a2}}$. Assume various values of p_{a2}, get corresponding values of p_{s2}, and from the superheat table observe the value of p_{s2} for which the relation $v_2 = k T$ is satisfied.

Having p_{a2} and p_{s2}, the energies U_1 and U_2 and the entropies S_1 and S_2 are readily found from the general formulas; then $u_2 - u_1$, Q, and W are calculated as in the preceding case.

14. *Work of Air Cycle on Rankine Principle.*—Although the theoretical Rankine cycle is based on steam as the working medium, it seems possible and perfectly proper to develop an air cycle or an air-steam cycle undergoing the same thermodynamic changes as those shown by the Rankine diagrams. These changes, illustrated by the $P-V$ diagram of Fig. 14 and the $T-S$ diagram of Fig. 15, consist of, first, adiabatic expansion from point (1) to point (2); second, exhaust to the atmosphere at constant pressure and constant temperature; third, admission of a new charge of the working medium,

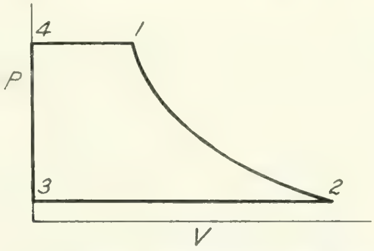

FIG. 14. P–V DIAGRAM FOR RANKINE CYCLE, COMPLETE EXPANSION TO BACK PRESSURE

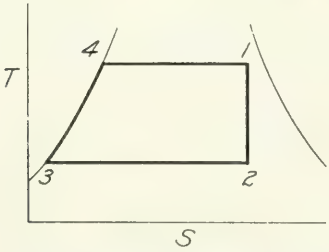

FIG. 15. T–S DIAGRAM FOR RANKINE CYCLE, COMPLETE EXPANSION TO BACK PRESSURE

which, because of lack of clearance in the theoretical engine causes a rise of pressure at constant volume followed by an increase in volume at constant pressure to the point of cut-off (1); thus the cycle is completed.

The work done during the cycle in which expansion continues to the back pressure is equal to the heat put in up to the point of cut-off minus the heat rejected after the adiabatic expansion ceases, that is, algebraically

$$AW = i_1 - i_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

If the expansion is *not* complete, as represented by the diagrams of Figs. 16 and 17, it is necessary to use a different expression for the work of the cycle. To develop such an expression, consider two cycles, one in which adiabatic expansion takes place from point (1) to point (3), and the other in which adiabatic expansion starts at point (2) and continues to point (3). It will be noted that both of these assumed cycles expand to the back pressure; hence equation (40) is applicable to each. In the first case, the work of cycle 1–3–4–5–7 is

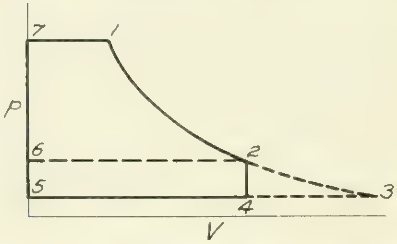

FIG. 16. P–V DIAGRAM FOR RANKINE CYCLE, INCOMPLETE EXPANSION; RELEASE PRESSURE HIGHER THAN BACK PRESSURE

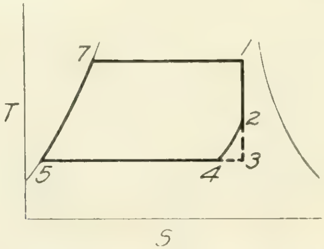

FIG. 17. T–S DIAGRAM FOR RANKINE CYCLE, INCOMPLETE EXPANSION; RELEASE PRESSURE HIGHER THAN BACK PRESSURE

$(i_1 - i_3)$. In the second case, the work of cycle 2–3–4–5–6 is $(i_2 - i_3)$. The work represented by the area 2–4–5–6, common to both cases, is $A(p_2 - p_3)v_2$ expressed in B.t.u. It is evident then that the work represented by area 1–2–4–5–7 is

$$\begin{aligned} AW &= (i_1 - i_3) - (i_2 - i_3) + A(p_2 - p_3)v_2 \\ &= i_1 - i_2 + A(p_2 - p_3)v_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41) \end{aligned}$$

15. *Specific Heat of Air.*—Several attempts have been made to establish the variation of the specific heat of air with temperature and pressure, but the results available at the present time are to a great extent discordant and contradictory. Probably the most reliable values of the specific heat at ordinary temperatures are those published by Swann* in 1910. His results, in terms of the 20-degree calorie are as follows:

$c_p = 0.24173$ cal. per gram-degree at 20 degrees centigrade;
$c_p = 0.24301$ cal. per gram-degree at 100 degrees centigrade.

In terms of the mean, (17½-degree) calorie, these values reduce to 0.2415 at 68 degrees Fahrenheit and 0.2428 at 212 degrees Fahrenheit.

The most recent data were published by Holborn and Jakob† in 1914. The tests covered the pressure range from 1 to 200 atmospheres, and at a mean temperature of 60 degrees centigrade. The results show a decided increase of c_p with pressure, but the result obtained at atmospheric pressure is in close agreement with the values given by Swann. For 140 degrees Fahrenheit and atmospheric pressure, $c_p = 0.2415$, while at 25 atmospheres $c_p = 0.2490$.

Scheel and Heuse found that at 20 degrees centigrade and atmospheric pressure $c_p = 0.2410$, but at lower temperatures they found larger values, a result contrary to general belief.

Owing to lack of sufficient and uniform data with which to deduce the variations in question, and owing to the fact that these variations appear to be small over the range of pressures and temperatures encountered in this investigation, the specific heat of air is considered as constant and equal to 0.242.

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* "On the Specific Heats of Air and Carbon Dioxide at Atmospheric Pressure by the Continuous Electric Method at 20° C. and 100° C.," Phil. Trans. Roy. Soc. of London, v. 210; p. 199, 1910.

† Holborn, L. and Jakob, M. "Die spezifische Wärme c_p der Luft bei 60° C. und 1 bis 200 t., Zeitschrift des Vereines-deutscher Ingenieurs, v. 58, p. 1429, Oct. 3, 1914.

IV. Construction and Use of Charts

16. *Heat Content—Entropy Charts.*—In the pocket attached to the back cover page are contained eight charts which illustrate the thermal properties of various mixtures of air and steam, and which were constructed from data obtained according to the following scheme. Dalton's Law states that when two or more gases and vapors occupy a space in common, each behaves as though the others were absent, and the pressure upon the walls enclosing the space or at any point within the space must be the sum of the pressures exerted by all the constituents of the mixture. After this well-known law is applied to a mixture of air and steam, it is apparent that each substance occupies the total volume and exerts a certain definite pressure, known as its partial pressure, such that the sum of the partial pressures is equal to the total pressure of the mixture. For saturated steam the pressure is fixed by the temperature, or vice versa, and determined from the steam tables; likewise the other properties, such as specific volume, heat content, entropy, etc., are determined. It is assumed that Boyle's Law for a perfect gas holds for air; whence

$$\frac{P_a V}{T} = 53.34$$

It was found to be most covenient to keep the air constituent equal to one pound for all mixtures; hence all the charts are based on one pound of air, and the weight of steam in each case is fixed by the percentage of steam under consideration.

As a preliminary to further proceedings, a table of values common to all mixtures was made, in which the main headings of columns were "pressure of mixture," (p_m), and "temperature of mixture," (t), respectively. The pressures ranged from atmospheric pressure to 150 pounds per square inch absolute, and the temperature extended from 32 degrees to the point where the partial pressure of the steam reached the pressure of the mixture, which for a pressure of 150 pounds was between 355 and 360 degrees Fahrenheit. G. A. Goodenough's latest steam tables were used. For each temperature the corresponding steam pressure, (p_s), specific volume, (v_s), heat content, (i''), heat of liquid, (i'), entropy of the steam, (s''), and entropy of the water, (s') were found and tabulated opposite the temperature. Since all these quantities are for one pound of steam, Boyle's Law and Dalton's Law were applied consecutively to determine the maximum weight of dry saturated steam which could exist with the air under each condition of temperature and pressure. With the partial pressure of the air, (p_a), which is the difference between the pressure of the mixture and the partial pressure of the steam, and with the absolute temperature and the gas-constant, the volume occupied by the air is readily computed. Since the steam must occupy the same volume, its weight equals the volume divided by the specific volume. This weight is designated by M_{sa} and is the weight of steam or vapor present with the air when the space is saturated. In any case, where the mixture under consideration contains more than the amount of steam necesssary for saturation under the prevailing conditions of temperature and pressure, the excess steam is regarded as water.

To find the values for entropy of air the following formula, derived in G. A. Goodenough's "Principles of Thermodynamics," is employed:

$$s_2 - s_1 = c_p \log_e \frac{T_2}{T_1} - AB \log_e \frac{p_2}{p_1}$$

in which

$s_2 - s_1$ represents the "change of entropy" (commonly called "entropy") between points 1 and 2, point 1 usually being an arbitrary reference point;

c_p is the specific heat of air at constant pressure, numerical value of which the theoretical discussion of Part III shows to be somewhat in question, but which is assumed constant and equal to 0.242 for the range of temperatures and pressures here considered;

T_1 and T_2 are absolute temperatures at point 1 and point 2, respectively, while p_1 and p_2 are absolute pressures at the same points;

AB is a constant, being the product of the gas-constant and the reciprocal of Joule's Equivalent, or $53.34 \div 777.64$.

With the arbitrary point 1 at T_1 taken as 1 degree absolute temperature and p_1 taken as 1 pound absolute pressure, the formula becomes very simple to use. Substituting the numerical values of constants and using 2.3025851 $\log_{10}$ in place of $\log_e$ gives the formula as finally used:

$$s_a = 0.55722559 \log_{10} T - 0.15793926 \log_{10} p_a$$

The entropy of water as tabulated in the steam tables is above 32 degrees Fahrenheit; consequently the entropy of steam, which is the sum of the entropy of evaporation, $\frac{r}{T}$, plus the entropy of water, is also the entropy above 32 degrees. Since the entropy of the air is taken with reference to a datum point different from that of the steam the resultant datum point for entropy of the mixture is imaginary, but this fact does not detract from the usefulness of the charts.

The charts are divided into two distinct regions; namely, the saturation region, extending below the saturation line, and the superheat region, extending above the saturation line. In the saturation region water is present with the steam and the air; and the heat content of the mixture, or the entropy of the mixture, is determined by the sum of the individual heat contents, or the individual entropies, of the three constituents. In the superheat region, however, there are only two constituents since no water is present, and the weight of steam is constant for a given mixture.

The method of determining the heat content and the entropy of the steam in the superheat region involves treatment somewhat different from that previously explained for saturated steam. As a starting point, values at regular intervals were assumed for the partial pressure of the steam. For each assumption at the temperature under consideration the specific volume, the heat content, and the entropy per pound are fixed by the steam tables. From the specific volume and the weight of steam the actual volume occupied by the steam is found by multiplication. This result being the volume of the air as well as that of the steam, it becomes possible to find the partial pressure of the air from Boyle's Law. Adding the partial pressure of the air to the partial pressure of the steam originally assumed gives the total pressure of the mixture. The heat content and the entropy of the steam in the mixture are determined by the weight of steam. Values for pressures intermediate to those assumed were found by graphical interpolation.

The quality lines shown in the saturation region were determined in each case from a set of constant pressure curves plotted between M_{sa} (weight of dry saturated steam corresponding to temperature) and i_m (heat content of mixture), the weights of dry saturated steam at various qualities being those percentages of the total weight of steam in the mixture M_s. The heat contents thus found for the various qualities were plotted on the corresponding pressure lines of the charts, and the quality lines drawn as shown.

The volume lines were determined in a somewhat similar fashion, except that in each case a set of constant temperature curves were plotted between V_m (volume of mixture) and s_m (entropy of mixture), and then for various volumes the entropy corresponding to each of the various temperatures was transferred from these curves to the chart.

17. *Application of Charts.*—The analytical solution of problems connected with mixtures of air and steam entails considerable complication and no small amount of calculation. The heat content—entropy charts presented herewith may be used to simplify the solution of such problems in much the same way as a Mollier steam chart facilitates the handling of steam problems.

The charts enable one to anticipate the resultant conditions of a mixture when the initial conditions of the air and of the steam are known, provided one condition of the mixture is known or assumed. They also facilitate the solution of problems wherein a mixture of air and steam undergoes adiabatic or isothermal expansion or compression. The solution of a few typical problems will illustrate their application more clearly.

Problem 1.

Let it be required to find the state of a mixture resulting from the combination of one pound of air at 114.5 pounds per square inch absolute pressure and 100 degrees Fahrenheit initial temperature, with 0.111111 pounds (10 per cent) of dry saturated steam at 114.5 pounds per square inch absolute pressure; the pressure of the mixture also to be 114.5 pounds per square inch absolute.

In solving this problem, first find the heat content of the mixture by adding the individual heat contents of the air and of the steam before mixing. That for the air is given by the expression

$$i_a = M_a \, c_p(t - 32)$$

in which c_p should be taken as 0.242. By substitution,

$$i_a = 1 \times 0.242 \times (100 - 32) = 16.45 \text{ B.t.u.}$$

The heat content of the steam should be determined by the use of G. A. Goodenough's latest steam tables. For the given pressure, $i'' = 1190.6$ B.t.u. per pound.

$$\begin{aligned} i_s &= M_s \; i'' \\ &= 0.111111 \times 1190.6 = 132.29 \text{ B.t.u.} \end{aligned}$$

Adding, $i_m = i_a + i_s$

$$= 16.45 + 132.29 = 148.74 \text{ B.t.u.}$$

Referring now to the chart for 10 per cent steam, find the intersection of the horizontal heat content line, 148.74, with the absolute pressure line, 114.5. This point represents the state of the resultant mixture, the temperature of which appears to be 210 degrees Fahrenheit, the quality 0.794, and the volume about 2.46 cubic feet.

The volume may be determined with somewhat greater precision by the use of steam tables. Since the volume of water in ordinary mixtures is inappreciable, the volume of the mixture is practically equal to the volume of dry steam in the mixture, as given by the relation

$$V = M_s \; x_s \; v''$$

The specific volume of dry saturated steam corresponding to a temperature of 210 degrees Fahrenheit is 27.83 cubic feet per pound. By substitution,

$$V = 0.111111 \times 0.794 \times 27.83 = 2.457 \text{ cu. ft.}$$

Problem 2.

Let it be required to find the resultant state of the mixture of Problem 1 after adiabatic expansion to four times its initial volume.

The final volume obviously equals 9.83 cubic feet. Refer to the chart again, and note the intersection of this volume line with the adiabatic. The temperature at this point is 115.6 degrees Fahrenheit, the quality 0.388, the pressure 23 pounds per square inch absolute, and the heat content 74 B.t.u.

Problem 3.

Let it be required to find the amount of work done during the adiabatic expansion of Problem 2.

In general, work done is equal to the change of intrinsic energy,

or $$W = U_1 - U_2$$

Also, by definition, $i = u + Apv$, whence

$$AU_1 = i_1 - 144\,A\,p_1v_1 = i_1 - 0.1852\,p_1v_1$$
$$AU_2 = i_2 - 144\,A\,p_2v_2 = i_2 - 0.1852\,p_2v_2$$

By substitution of the values previously determined,

$$AU_1 = 148.74 - 0.1852 \times 114.5 \times 2.457 = 96.64 \text{ B. t. u.}$$
$$AU_2 = 74.0 - 0.1852 \times 23 \times 9.83 = 32.1 \text{ B. t. u.}$$
$$AW = AU_1 - AU_2 = 96.64 - 32.1 = 64.54 \text{ B. t. u.}$$
$$W = 778\,AW = 778 \times 64.54 = 50\ 212 \text{ foot-pounds.}$$

18. *Saturation Diagram.*—The Saturation Chart contained in the pocket on the back cover page, which chart is constructed for various pressures with percentages of steam as ordinates and temperatures as abscissæ, was determined by the combined use of G. A. Goodenough's tables for saturated steam and the perfect gas equation for air. For mixtures of 50 per cent steam or less, the diagram shows both the relation between total pressure and temperature of mixture, and the corresponding volumes and partial pressures when the steam in the mixture is dry and saturated. It is probable that this condition prevails in all engine tests of mixtures of air and steam except those in which the air was supplied at a temperature of something over 200 degrees, or when the percentage of steam was very small. The so-called mixing chamber used in the engine tests was a very effective separator, and it seems safe to assume that a wet mixture becomes practically freed of entrained moisture upon leaving this part of the apparatus. In order to prove the validity of this assumption, the curves shown in Figs. 18, 19, and 20 were plotted with observed temperatures as ordinates and theoretical saturation temperatures as abscissæ. These curves show very clearly where the steam in the mixture was saturated, where it was superheated, and the amount of superheat. When the possibility of experimental errors is considered, the agreement of the observed with the theoretical temperatures where the saturated condition prevails is in general satisfactory.

Because of the close agreement between the actual and theoretical saturation temperatures of air-steam mixtures, it is possible that the injection of steam into air until it is known to be saturated with water vapor, and the subsequent determination of the steam added through its condensation, may be an accurate method of determining the

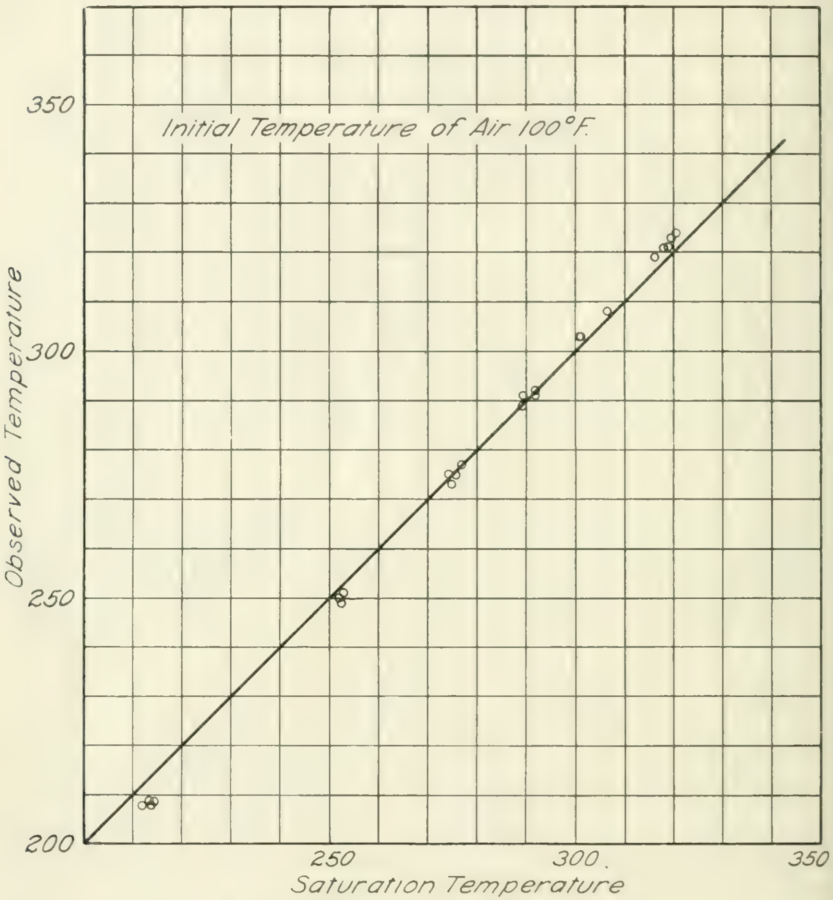

FIG. 18. COMPARISON OF OBSERVED TEMPERATURES WITH COMPUTED SATURATION TEMPERATURES

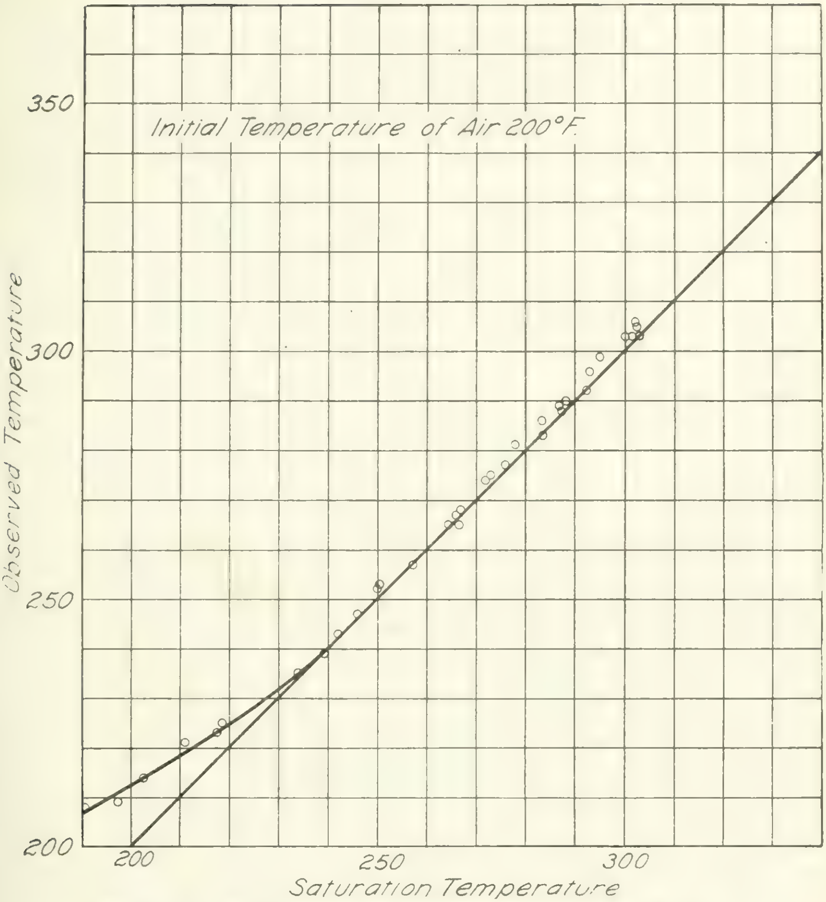

FIG. 19. COMPARISON OF OBSERVED TEMPERATURES WITH COMPUTED SATURATION TEMPERATURES

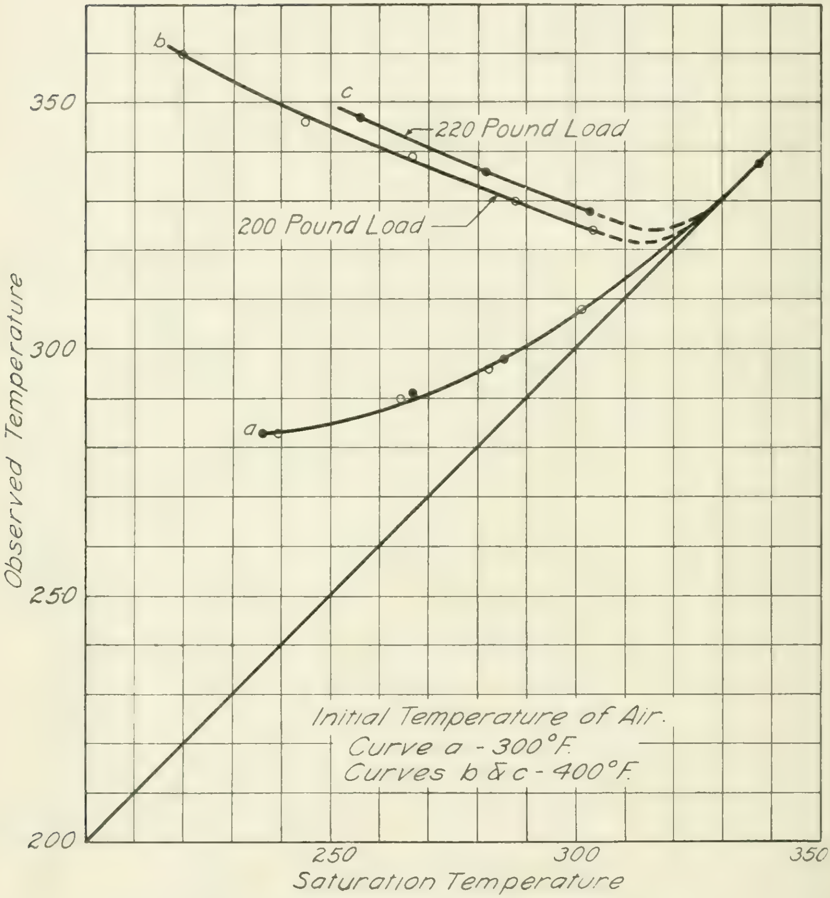

FIG. 20. COMPARISON OF OBSERVED TEMPERATURES WITH COMPUTED SATURATION TEMPERATURES

weight of air in the mixture. If this method is employed it is advisable to use the air cold, and to add enough steam to ensure saturation in the mixture. This condition may easily be ascertained by the use of a separator; furthermore the separator may be relied upon to remove practically all the entrained moisture. Pressure and temperature readings should be ascertained directly after the mixture of air and dry saturated steam leaves the separator. The temperature must be ascertained with precision; so it is essential that the thermometer be carefully calibrated and used bare rather than in a thermometer-cup, and that the necessary "stem-correction" be applied. In order to determine the amount of steam in the mixture a surface condenser must be used, and the mixture cooled as much as possible. The temperature of the air leaving the condenser is a measure of the amount of moisture it carries off, which should be computed and added to the condensate. The lower this exhaust temperature, the less will be the amount of moisture thus lost.

19. *Theoretical Gain from Mixing Air and Steam, Determined by Aid of Charts.*—That quantities of air and steam are capable of doing more work expansively when mixed than equal quantities of the two media acting separately can accomplish is evidenced to a startling degree by the results of the tests. As an explanation of this peculiar phenomenon Reynolds' experiments have been cited as evidence that the cylinder condensation is greatly reduced by the presence of air. Importance is also attached to Rankine's theory that the gain from mixing is partly due to increasing the working temperature limits of the steam. The specific heat of air is comparatively low, and expansion of air against a piston is therefore accompanied by a very rapid drop in temperature; thus the extremely low temperatures experienced in tests with cold air as the working medium are explained. The fall in temperature is accompanied by a correspondingly rapid decrease in pressure, with the result that the effective force against the piston and the net work done are comparatively low.

By the addition of a small amount of steam, with its latent heat of vaporization available for contribution to the heat content of the air during expansion, the rate of temperature decrease is lessened and the mean pressure of the air during its expansion is very much higher than when working alone. To show the effect of this action a theoretical pressure-volume (Rankine) diagram, Fig. 21, has been

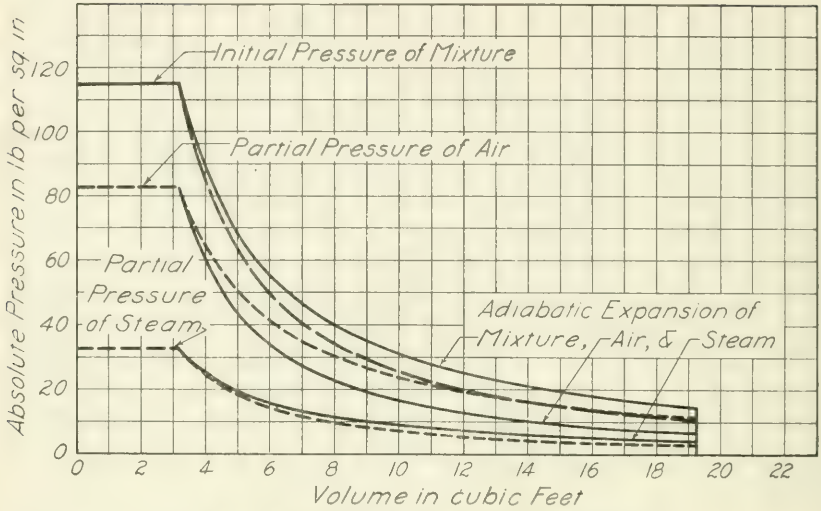

FIG. 21. P-V DIAGRAM FOR AIR-STEAM MIXTURES

drawn for a saturated mixture containing one pound of air and 20 per cent of steam with adiabatic expansion from an initial pressure of 115 pounds per square inch absolute.

Data for the adiabatic (isentropic) expansion of the mixture were procured from the 20 per cent chart, as were also the temperatures of the mixture at various stages of the expansion. Corresponding to these temperatures are certain pressures for saturated steam, shown by steam tables, and these pressures are the partial pressures of the steam in the mixture. Subtracting the partial pressure of the steam at various points from the corresponding pressure of the mixture gives the partial pressures of the air. The partial pressures of the steam and of the air during the adiabatic expansion of the mixture are represented by the dotted lines. The full line adjacent to each dotted line represents adiabatic expansion of the constituent, that for air being determined by means of the gas equation $p_1v_1{}^k = p_2v_2{}^k$, while the adiabatic for steam was computed in connection with the Mollier diagram.

The dotted line which represents the actual change of condition of the air in the mixture naturally lies above the adiabatic for air, because the air receives heat from the steam during the process of expansion of the mixture. The area between these two curves is a measure of the work gained from the air by virtue of the heat acquired from the steam during the adiabatic expansion of the mixture. On the other hand the dotted line representing the variation in partial pressure of the steam during adiabatic expansion of the mixture is below the adiabatic for steam alone, and the area between these two curves represents the energy given to the air by the steam. The area showing an increase of work from the air is much larger than the area representing the decrease in work done by the steam, the difference obviously being the net gain due to mixing.

Summing up the pressures of the air and of the steam when each expands adiabatically gives the dashed line lying near the curve for adiabatic expansion of the mixture. The area between these two lines is exactly equal to the difference of the two areas mentioned in the preceding paragraph, and likewise represents the net gain due to mixing.

Other mixtures were investigated in the same manner and with similar results, but it was decided that additional diagrams of this sort would be of no particular value, since the theoretical gain with various mixtures is shown in a more vivid and comprehensive manner by the curves of Figs. 22 and 23.

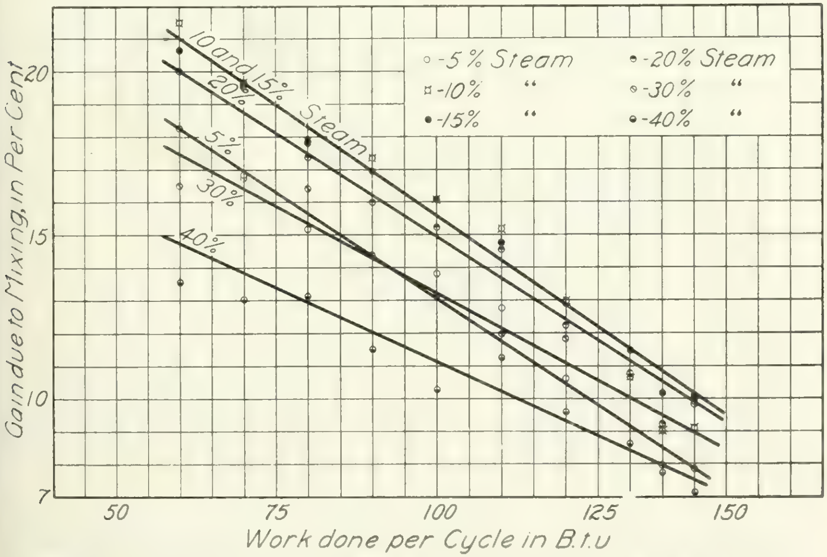

FIG. 22. THEORETICAL GAIN IN PER CENT DUE TO MIXING FOR DIFFERENT AMOUNTS OF WORK DONE PER CYCLE

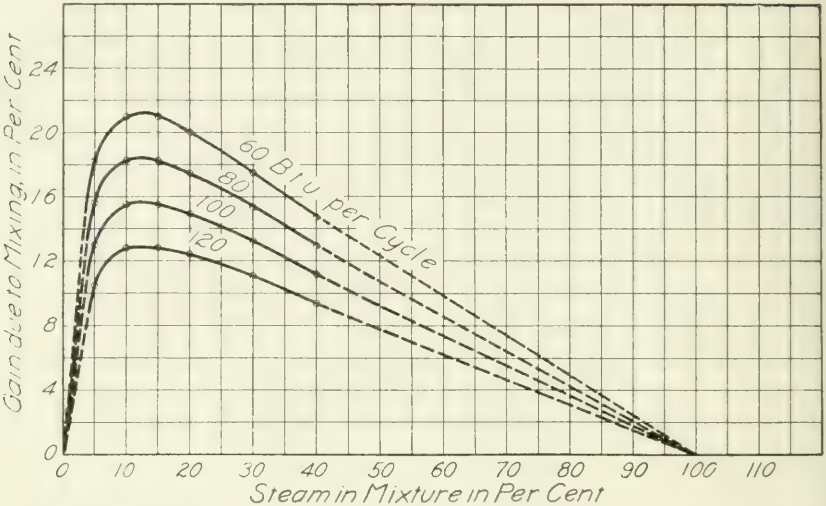

FIG. 23. THEORETICAL GAIN IN PER CENT DUE TO MIXING FOR DIFFERENT PERCENTAGES OF STEAM IN MIXTURE

The data for these curves were obtained in the following manner. It was assumed that an engine operated on a cycle similar to the theoretical Rankine cycle, at variable cut-off, with an initial pressure of 115 pounds per square inch absolute, and exhausting against atmospheric back pressure of 14.7 pounds per square inch. The problem was then one of determining the consumption at different loads:

(1) when working with air alone at given initial temperatures;

(2) when working with dry saturated steam only;

(3) when working with different saturated mixtures of air and steam.

After these determinations were made and the results plotted, as shown in Figs. 24 and 25, the gain due to mixing was found and plotted in exactly the same manner as previously described for the experimental data.

The determination of the theoretical air consumption at various loads was made by taking different weights of air and by computing the work that each would do in accordance with the Rankine diagram.

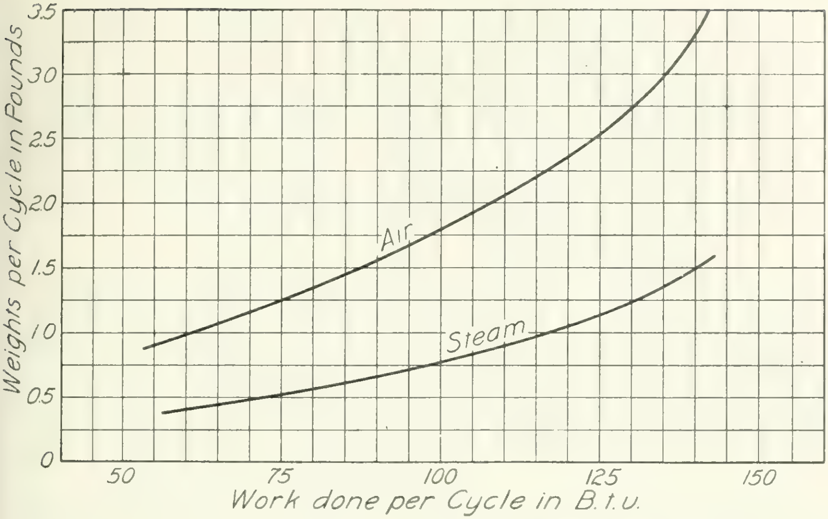

FIG. 24. THEORETICAL CONSUMPTION OF AIR AND STEAM RESPECTIVELY PER CYCLE FOR DIFFERENT AMOUNTS OF WORK DONE PER CYCLE

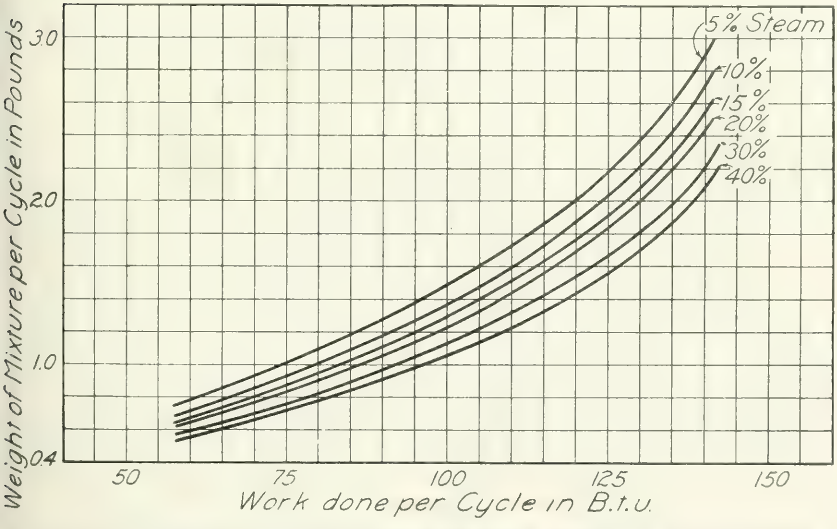

FIG. 25. THEORETICAL CONSUMPTION OF MIXTURE PER CYCLE FOR DIFFERENT AMOUNTS OF WORK DONE PER CYCLE AND DIFFERENT PERCENTAGES OF STEAM IN MIXTURE

The work of the cycle, as given by the general equation (41), is

$$AW = i_1 - i_2 + A(p_2 - p_3)\, v_2$$

The final volume for all cases was arbitrarily chosen as 7.85 cubic feet, the volume of one pound of air at the end of the adiabatic expansion. Substituting this value of v_2, together with the constants previously given, the equation reduces to the simple form

$$AW = i_1 - i_2 + 1.454\ (p_2 - 14.7)$$

The initial volume of one pound of air, at the initial pressure of 115 pounds per square inch absolute and with an initial temperature of 100 degrees Fahrenheit, was found to be equal to 1.82 cubic feet; therefore, the values of v_1 for the different assumed weights were easily found by multiplication. The number of expansions in each case was then determined by taking the ratio $\frac{v_2}{v_1}$. The ratio times the volume of one pound of air at the initial pressure gives the volume to which the working substance would have to expand at the different ratios of expansion, provided exactly one pound of air was used in every case. The purpose of making this computation was to enable one to use C. R. Richards' "Entropy-Temperature Diagram for one Pound of Air"* for determining the final pressures and temperatures for each of the assumed conditions. Finally the heat content of air is expressed by the general equation

$$i_a = M_a\, c_p\, T \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (42)$$

hence $i_1 - i_2 = M_a\, c_p\ (t_1 - t_2)$. With the initial temperature taken as 100 degrees Fahrenheit and c_p as 0.242, the equation for this set of computations reduced to $i_1 - i_2 = 0.242\ M_a\ (100 - t_2)$.

Theoretical steam consumption data were determined in a similar manner by the aid of a Mollier diagram, with the same general formula for the work of the cycle. The same scheme was applied in the case of different mixtures after the state of the mixture was determined, it being assumed in each case that one pound of air at 100 degrees Fahrenheit was mixed with the given percentage of dry saturated steam. By adding the heat contents of the constituents before mixing the heat content of the mixture was obtained; then by use of the chart the initial state of the mixture was determined. For each different percentage the final volume was found for each assumed ratio of expansion when the weight used in every case was the same as the unit weight upon which the chart was based. By the aid of the charts it was a simple matter to determine the physical and thermal properties of the mixture for any degree of expansion, that is, for any ratio of expansion, since adiabatic expansion is a constant entropy process. Each chart is based upon a definite weight of mixture which in every case contains just one pound of air; hence the total weights for the several charts are different. This fact must always be borne in mind when the charts are used.

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* Univ. of Ill. Eng. Exp. Sta., Bul. 63, 1913.

Comparison of Fig. 23, which represents the theoretical gain from mixing when working on the theoretical Rankine cycle without clearance and with perfect adiabatic expansion of the mixture, with Fig. 6, which shows the actual gain in a real engine, brings out the fact that there is a striking similarity between the two. There is some difference in the percentage of steam which gives the greater gain, it being between 10 and 15 per cent in the theoretical case and about 25 per cent in the tests. The maximum gain is a function of the load, and as nearly as can be determined from both the theoretical and the test results it is approximately a linear function.

The curves of Fig. 26 are produced for each of the working sub-

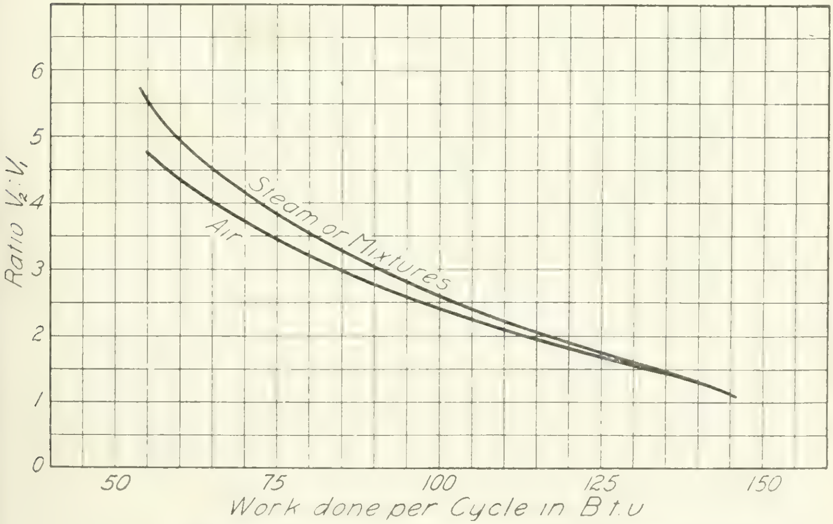

Fig. 26. Relation Between Expansion Ratio and Work Done per Cycle for Air, Steam, and Mixture

stances in order to give some idea of the relation of the work done per cycle, as expressed in B.t.u., to the cut-off or ratio between initial and final volumes. This relation is practically the same for steam and all the various mixtures, as is evident from the fact that a single curve approximates all plotted points for steam and mixtures. The curve for air, however, lies below the curve for steam and mixtures; thus it is suggested that a greater volume of air is required to do the same work as can be done by a given volume of steam or air-steam mixture. This indication is in harmony with the test results. Examination of the indicator cards taken during the mixture tests with air at 100 degrees Fahrenheit shows that a 150-pound brake load corresponds to a cut-off at or near 14 per cent of the stroke; a 175-pound load to a cut-off at or near 18 per cent; a 200-pound load to a cut-off at or near 22 per cent, and a 225-pound load to a cut-off at or near 26 per cent.

Computations were not made with air at other temperatures, but it is very probable that the same sort of agreement would be found between theoretical gain and actual gain for every case.

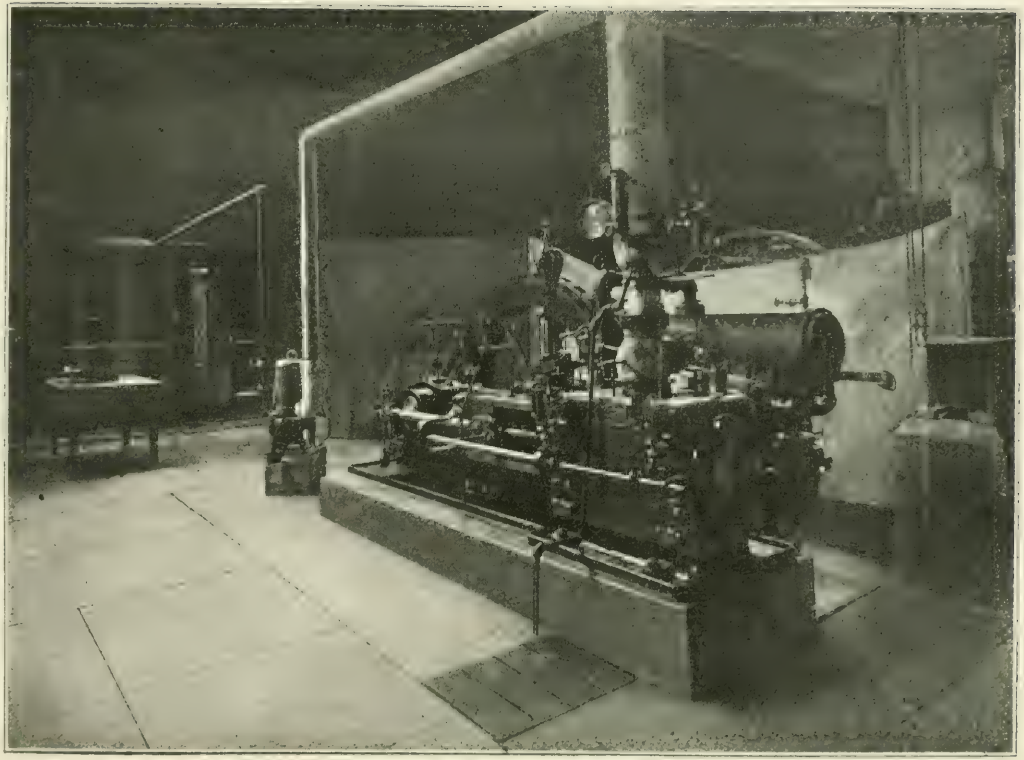

FIG. 27. GENERAL ARRANGEMENT OF EXPERIMENTAL PLANT

APPENDIX A

DESCRIPTION OF EXPERIMENTAL PLANT

The photograph, Fig. 27, taken in connection with the tests of reheated air previously mentioned, shows substantially the arrangement of equipment used in the present investigation except that a reheater of different design was employed in place of the one shown. A diagrammatic arrangment of the equipment used is shown in Fig. 28. The air-compressor is not shown in either figure but its use is explained and its relative position indicated in the following description of the plant. To reduce the pulsations of pressure from the compressor, the air was delivered to the first receiver shown in Fig. 28. From the first receiver the air passed through the Venturi tube to a second receiver employed to reduce the pulsations of pressure produced in the air pipe by the engine.* Between the second receiver and the engine the air passed through several feet of pipe which could be heated externally by gas burners. When the air was heated to the desired temperature it was delivered into a mixing chamber located immediately above the engine throttle valve. High pressure steam could also be admitted to this mixing chamber, as shown in the figure. By this arrangement it was possible to secure the desired proportion of steam and air for the mixture tests.

The engine was built by the Murray Iron Works Company of Burlington, Iowa. The builders' rating for steam when operating non-condensing with 100 pounds gage pressure, 100 r. p. m., and cut-off at one-quarter stroke, is 24 indicated horse power. The governor is of the weighted fly-ball type. The principal dimensions of the engine are as follows:

Cylinder diameter	8.0	inches
Stroke	18.05	"
Piston-rod diameter	1.875	"
Shaft diameter	4.0	"
Shaft length	66.0	"
Main bearing diameter	4.0	"
Main bearing length	10.0	"

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* Wilson, L. A. "Action of Venturi Meter with Pulsating Flow." Power, v. 44, p. 425, Sept. 19, 1916.

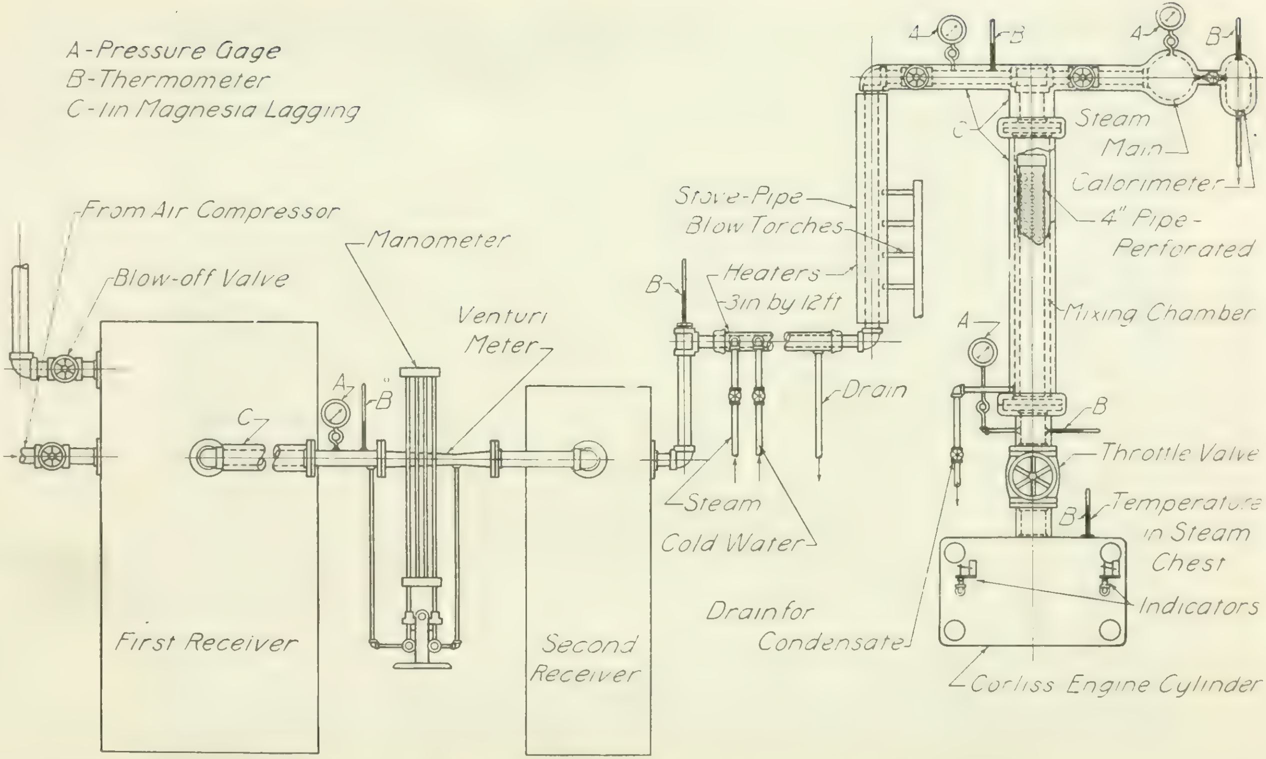

FIG. 28. DIAGRAMMATIC ARRANGEMENT OF APPARATUS

Outer bearing diameter	4.0 inches
Outer bearing length	10.0 "
Fly-wheel diameter	6 feet
Fly-wheel face	9.0 inches
Fly-wheel weight	3100 pounds
Crank-pin diameter	2.25 inches
Crank-pin length	2.75 "
Crosshead pin diameter	2.25 "
Crosshead pin length	2.75 "
Crosshead shoes width	4.75 "
Crosshead shoes length	10.0 "
Steam pipe diameter	2.5 "
Exhaust pipe diameter	3.0 "
Length of engine from center of shaft to back end of cylinder plate	8 feet, 3.5 inches
Head-end clearance	5.48 per cent piston displacement
Crank-end clearance	5.63 per cent piston displacement

The accessories of the engine used in all tests consisted of a Prony brake, condenser, weighing tank, revolution counter, reducing motion, mixing chamber, calorimeter, and two Buffalo scales, one used with the brake and the other for weighing the condensate. The condenser was of the Dean surface type having 100 square feet of condensing surface. There was no vacuum pump attached. A thermometer cup was inserted at the outlet of the condenser for taking the temperature of the condensate and air. The reducing motion was of the "rod and groove" type with grooved and notched discs for engaging the indicator cord. The discs were $2\frac{3}{4}$ inches in diameter, and the diagrams produced were approximately 4 inches long. An adjustable force-feed lubricator furnished oil to the cylinder and valves.

The air compressor was an Ingersoll-Sergeant two-stage cross-compound machine having a maximum capacity of approximately 375 cubic feet of free air per minute at a pressure of 100 pounds per square inch gage. The cylinder dimensions of the compressor were as follows:

	Inches
Diameter high-pressure steam cylinder	12.0
Diameter low-pressure steam cylinder	22.0
Diameter low-pressure air cylinder	18.25
Diameter high-pressure air cylinder	12.25
Stroke	12.0

The Venturi meter, made by the Builders' Iron Foundry of Providence, R. I., had an up-stream diameter of 2½ inches and a throat diameter of ¾ inch; thus the ratio of contraction was 0.3. It was set in a 2½-inch pipe line between two receivers, 7 feet from the nearest up-stream bend and 5 feet from the nearest up-stream obstruction in the pipe. The obstruction was a thermometer cup which held a thermometer to indicate the temperature of the air; from that point to the meter the pipe line was lagged. The meter was in three sections. The first was the inlet section, 7⅜ inches long, having a pressure chamber with pipe connection leading to one side of the manometer. A gage was also attached to this chamber. The middle section, 10⅜ inches long, contained the throat-pressure chamber which was connected to the other side of the manometer. The third section was an expanding nozzle, 12⅜ inches long, through which the increase in velocity head acquired at the throat was converted back to pressure head. The manometer had a sliding scale, 22 inches long, divided into tenths of an inch. There were pockets at the bottom of the manometer fitted with needle valves to drain off any mercury or water which might accidentally be blown over.

The receiver between the compressor and the meter had a capacity of 65 cubic feet. It was fitted with a blow-off valve which was adjusted by means of a sliding weight. The second receiver, which had a capacity of 17 cubic feet, was located between the meter and the engine. The meter was thereby freed from the disturbing effect of the reciprocations of both engine and compressor, so that there was no appreciable oscillation of the liquid in the manometer.

A portion of the two-inch pipe leading from the second receiver to the engine was fitted with a jacket composed of 12 feet of 3-inch pipe sealed with an ammonia fitting at each end. Both cold water and steam were piped to the jacket, and by this arrangement it was possible to regulate the temperature of the air within the lower range, the highest temperature thus attained being slightly greater than 200 degrees Fahrenheit. For higher temperatures a battery of four blow-torches was arranged to play on a vertical section of the pipe which was surrounded by approximately four feet of stove pipe covered with asbestos lagging. With both heaters utilized simultaneously a temperature of 400 degrees was readily obtained for the maximum amount of air used in the air tests.

The mixing chamber located immediately above the engine consisted of $4\frac{1}{2}$ feet of 6-inch pipe covered with magnesia lagging. A 4-inch perforated pipe closed at the end projected up into this mixer and caused any condensed steam present to be separated and held at the bottom of the chamber until drawn off at the end of the test.

Regulation of the steam pressure was effected by means of a throttle valve located in the steam main several feet from the engine and compressor. To make it possible for the operator of the valve to be near the engine a rope and pulley arrangement was used. The flow of air was regulated by a throttle valve placed between the first receiver and the Venturi meter.

A thermometer cup was inserted in the elbow where the air left the second receiver and entered the heater. It is probable that the air at that point was always saturated, and from the observed temperature the amount of water vapor carried by the air to the mixing chamber was computed.

APPENDIX B

CALIBRATION OF APPARATUS

The thermometers used in these tests were calibrated by comparison with standard thermometers having certificates of recent date from the Physikalisch-Technische Reichsanstalt or Bureau of Standards. A water bath heated by steam was used for temperatures below 200 degrees Fahrenheit. For higher temperatures an oil bath was used in which the oil was heated by means of an electric current and was vigorously stirred by a motor-driven agitator. Thermometers which proved to be in error more than half of a degree in the range over which they were to be used were kept for emergencies, and the most nearly accurate of the remainder were assigned to positions where accuracy was of the greatest importance.

The pressure gages were tested by means of a Crosby gage tester, and adjusted to read as nearly correctly as possible over the whole range, and at the same time to give exact readings at the pressures which they were to indicate during the tests. Standard test gages graduated to one pound were used to measure pressures at the throttle and at the up-stream side of the Venturi meter.

Tabor outside-spring indicators were used after being tested for accuracy in the usual manner. The springs were calibrated cold by means of the Crosby gage tester, the indicator itself being attached by special fittings. The springs were distinguished from each other by punch marks, and each spring was kept on the indicator with which it was calibrated. Sixty-pound springs were used throughout the tests.

The accuracy of the scales used with the Prony brake and of those for weighing condensate was checked by means of standard test weights. The tare due to the brake arm was found by balancing the scale beam when the brake wheel was turned slowly, first one way and then the other, and by averaging the net readings. A known weight used to hold down the brake arm was deducted from each reading. The tare of the brake plus the weight of the pedestal gave the total tare of the brake scales.

The Venturi meter used for measuring the air under pressure had been previously calibrated by John N. Vedder by comparison with a Durley gage box and standard orifices. Following the completion of this investigation the accuracy of the original method of calibrating the meter was questioned and it was therefore recalibrated by C. Z. Rosecrans during the fall of 1921, using the air weighing plant installed in connection with the calibration of air measuring devices employed in the research work on warm air furnaces and heating systems.* It was found that the original correction factor was 3 per cent too low. Fig. 29 shows the curve of correction factors obtained by Rosecrans to be applied to the results given by the regular Venturi meter formula for any value of W thus obtained. The formula in one of its various forms is

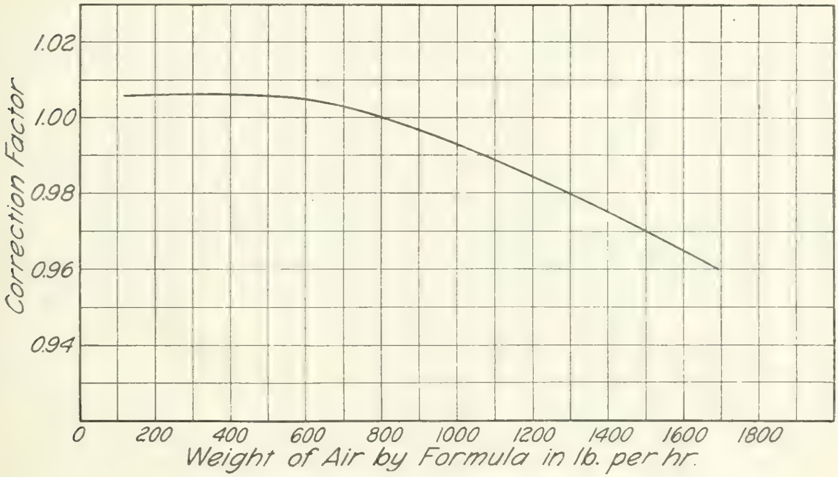

FIG. 29. CORRECTION FACTORS FOR VENTURI METER FORMULA

$$W = F_2\left(\delta_o \frac{T_o 2gk}{P_o(k-1)}\right)^{\frac{1}{2}} \frac{P_1}{T_1^{\frac{1}{2}}} \left(\frac{P_2}{P_1}\right)^{1/k} \left\{\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}}{1-\left(\frac{F_2}{F_1}\right)^2 \left(\frac{P_2}{P_1}\right)^{2/k}}\right\}^{\frac{1}{2}}$$

in which

W = weight of air per second flowing through the tube,
F_2 = area of throat of tube in square feet,
F_1 = area of up-stream section of tube in quare feet,

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* Univ. of Ill. Eng. Exp. Sta., Bul. No. 120, p. 81.

T_o = 492 degrees absolute in Fahrenheit units,
P_o = 2116 pounds per square foot,
g = acceleration due to gravity,
$k = \frac{c}{c_v}$ the ratio of the specific heats at constant pressure and constant volume,
δ_o = density of the air in pounds per cubic foot at P_o and T_o,
P_1 = pressure in pounds per square foot at entrance to the tube,
P_2 = pressure in pounds per square foot in the throat of tube,
T_1 = absolute temperature of air flowing, in degrees Fahrenheit.

The use of this formula involves laborious calculations which may be greatly simplified for a particular tube through the aid of graphical charts. G. B. Upton* simplified the formula without materially impairing its accuracy by expanding it according to the binomial theorem and dropping the smaller terms. Depending upon the accuracy desired three equations may thus be secured, as follows:

$$\text{(a)} \quad W = C_1 \left(\frac{P_1 \bigtriangleup P}{T_1}\right)^{\frac{1}{2}}$$

$$\text{(b)} \quad W = C_1 \left(\frac{P_1 \bigtriangleup P}{T_1}\right)^{\frac{1}{2}} \left[1 - C_2 \left(\frac{\bigtriangleup P}{P_1}\right)\right]$$

$$(c) \quad W = C_1 \left(\frac{P_1 \bigtriangleup P}{T_1}\right)^{\frac{1}{2}} \left[1 - C_2 \left(\frac{\bigtriangleup P}{P_1}\right) + C_3 \left(\frac{\bigtriangleup P}{P_1}\right)^2\right]$$

in which

$$\bigtriangleup P = P_1 - P_2$$

$$C_1 = F_2 \left\{ \frac{2g\delta_o\, T_o}{P_o\left[1 - \left(\frac{F_2}{F_1}\right)^2\right]} \right\}^{\frac{1}{2}}$$

$$C_2 = \frac{3 + \left(\frac{F_2}{F_1}\right)^2}{4k\left[1 - \left(\frac{F_2}{F_1}\right)^2\right]}$$

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* The Sibley Jour. Eng., v. 29, pp. 33 and 90.

$$C_3 = \frac{29 + 110\left(\frac{F_2}{F_1}\right)^2 + 5\left(\frac{F_2}{F_1}\right)^4 - 40k + 32\left(\frac{F_2}{F_1}\right)^2 k + 8\left(\frac{F_2}{F_1}\right)^4 k}{96k^2\left[1 - \left(\frac{F_2}{F_1}\right)^2\right]^2}$$

The relative accuracy of these three expressions is as follows:

Formula	Upper limiting value of $\frac{\triangle P}{P_1}$ for an error not exceeding	
	0.1 PER CENT	1.0 PER CENT
(a)	0.002	0.015
(b)	0.05	0.25
(c)	0.25	Never happens

APPENDIX C

NUMERICAL COMPUTATIONS

The following Venturi constants were used in the numerical computations:

Up-stream diameter = 2.5 inches
Throat diameter = 0.75 inch
Area of up-stream section = 0.0340882 square feet = F_1
Area of section at throat = 0.003068 square feet = F_2

$$\left(\frac{F_2}{F_1}\right)^2 = 0.0081003$$

$k = 1.405$
$2g = 64.32$
$\delta_o = 0.0807$ for dry air, and 0.0802 for saturated vapor mixture
$T_o = 492$ degrees absolute Fahrenheit
$P_o = 2116$ pounds per square foot (standard atmospheric pressure)

By substitution of these data in the equations for the Venturi meter constants, the numerical values were found to be:

$C_1 = 0.003374$ (considering saturated vapor mixture)
$C_2 = 0.53962$
$C_3 = -0.13915$

The portion of equation (c), Appendix B, involving the term $C_3\left(\frac{\triangle P}{P_1}\right)^2$ was in every case too small to be considered; so equation (b) was used except for cases where the value of $\triangle P$ was extremely small, when equation (a) applies without appreciable error.

When the readings of Venturi head in inches of mercury were reduced to equivalent pounds per square foot, it was found to be unnecessary to make corrections for slight changes in the temperature of the mercury, because the effect of this correction is inappreciable in the final results. For this reason the equivalent pressure of one inch of mercury at 62 degrees Fahrenheit, equal to 70.5 pounds per square foot, was used throughout the tests.

The engine constants used were as follows:

Diameter of piston = 8.0 inches,
Diameter of piston rod = 1.875 inches,
Piston area head end (A_1) = 50.2655 square inches,
Piston area crank end (A_2) = 47.5043 square inches,
Stroke (L) = 18.05 inches,
Brake arm length (b) = 5.25 feet,

$$\text{B.h.p.} = \frac{2\pi G b N}{33000} = 0.001\, G N,$$

$$\text{I.h.p.} = \frac{P L A N}{33000},$$

$$\text{I.h.p. head end} = 0.00229114\, P N,$$

$$\text{I.h.p. crank end} = 0.0021653\, P N;$$

in which

G = weight at end of brake arm in pounds,
N = revolutions per minute,
P = mean effective pressure.

APPENDIX D

HISTORY OF THE STUDY OF AIR-STEAM MIXTURES

Early Appreciation of Advantages of Air-Steam Mixtures

The use of air and steam combined as a medium for developing power is not of recent origin. The value of air in preventing rapid condensation of steam was known by Savery and other scientists before 1800 and was utilized by them in the operation of water pumps by interposing a layer of air instead of a piston between the steam and the water.

Reports of the United States Patent Office show that during the past century attempts have been made to patent an air-steam engine and that several patents have been granted. Very few, however, have been given prominence by means of discussion in engineering literature. George H. Babcock, in a paper entitled "Substitutes for Steam" presented to the American Society of Mechanical Engineers in 1885-6,[*] reviewed the various attempts to establish the use of this novel means of driving prime movers.

In 1872 W. J. M. Rankine[†] published an article entitled "Remarks on the Theory of Air and Steam Engines," in which he pointed out that the air has a tendency to increase the efficiency of the mixture, independently of any addition of heat, by widening the working temperature limits of the steam.

In 1873 Osborne Reynolds,[‡] Fellow of Queen's College, Cambridge, England, made an investigation to ascertain the effect of a small quantity of air on the power of a cold surface to condense steam. The apparatus employed consisted of a glass flask, in which a surface condenser was formed by a copper pipe passing in and out through the cork. The pipe, which was kept cool by a stream of water, was arranged so that all the condensate discharged from it in drops which could be counted. Reynolds reached the following conclusions:

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[*] Trans. A. S. M. E., v. 7, pp. 680 and 738, 1886.

[†] The Engr (London), v. 34, p. 293, Nov. 1; p. 348, Nov. 22; p. 381, Dec. 6; p. 431, Dec. 27; 1872.

[‡] Eng. (London), v. 16, p. 283, Oct. 3, 1873
Phil. Mag. v. 47, 1874.
Jour. Franklin Inst., v. 121, p. 401, June, 1886.

(1) A small quantity of air in steam greatly retards condensation upon a cold surface. On the other hand, there is no limit to the rate at which pure steam will condense except the power of the surface to carry off the heat.

(2) The rate of condensation diminishes rapidly and nearly uniformly as the pressure of the air increases from 2 to 10 per cent of that of the steam. It then diminishes less and less rapidly until a pressure of 30 per cent of that of the steam is reached, when the rate of condensation remains nearly constant.

(3) In consequence of this effect of air the necessary size of a surface condenser for a steam engine increases very rapidly with the quantity of air allowed within it.

(4) If air is mixed with steam before it is used, the condensation at the surface of a cylinder may be greatly diminished, and consequently the efficiency of the engine is increased.

(5) The maximum effect in diminishing condensation will be obtained when the pressure of the air is about one-tenth that of the steam, or when about 2 cubic feet of air at atmospheric pressure and with a temperature of 60 degrees Fahrenheit are mixed with each pound of steam.

The most potent cause of the reduced condensation of steam when mixed with air is the separation of the molecules of the steam by the molecules of the air. Evidently there can be no condensation of the steam unless its molecules coalesce; furthermore, the steam when mixed with air is converted into superheated steam, and therefore recedes from the condensation point.

According to Dr. A. Stodola,* the Physicist Pictet suggested the use of heated compressed air mixed with steam to lower the temperature drop of a non-condensing steam engine. Pictet concluded that the steam consumption of a non-condensing engine, operated with an air-steam mixture of about 2 to 1, would approach nearly that of a condensing engine.

Attempts to Use Air-Steam Mixtures in Engines

There appear to be two general classes of air-steam engines hitherto invented, depending upon the method of providing a mixture of air and steam. In one case the mixture is formed within the boiler, while in the other the air and steam are mixed at or near the engine. The first method involves such means as the following:

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* "Steam Turbines," 1905 edition, p. 410.

(1) Injecting the hot gases from the chimney or uptake into the boiler under pressure;

(2) Operating the furnace under a pressure sufficiently high to force the gaseous products of combustion into the boiler without further compression;

(3) Forcing air into the boiler to mix with the steam.

By any one of these means energy is required to pump or inject the air or gases into the high pressure steam. The device used for this purpose has been an injector, a pump connected to the engine, or a pump driven independently.

The first of the various attempts to use the products of combustion mixed with steam which received the attention of the engineering profession was an invention by Minus Ward* of Baltimore. Similar inventions appeared from time to time but no practical application was made until 1856, when a large locomotive constructed upon this principle was tried on the Erie Railroad. Owing to the deleterious effect of the ashes and intense heat on the cylinder walls of the engine, the test failed. The theoretical aspects of an engine of this design are interesting, and they show the efficiency which might be attained if it were possible to overcome the practical operating difficulties. The theoretical efficiency of such an engine was studied in detail by J. A. Henderson for his graduation thesis at Stevens Institute in 1873. The thesis was published by Professor Thurston in his report on "Mechanical Engineering at the Vienna Exposition in 1873," and printed in abridged form in the Journal of the Franklin Institute in 1874.† The following were Henderson's conclusions concerning the most advantageous use of mixed air and steam in engines utilizing the entire products of combustion.

(1) When a moderate value for the initial temperature of the mixture, t_1, is required in order to avoid making provision against the effects of heat, comparatively high initial pressure, p_1, and as small a supply of air, N, as possible are the most advan-

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* "Account of a New Steam and Heated Air Engine," Jour. Franklin Inst., v. 4, p. 49. 1827.

† Henderson, J. A. "The Theory of Aero-steam Engines," Jour. Franklin Inst., v. 98, pp. 17, 103, and 185, 1874.

tageous conditions for yielding the highest efficiency consistent with the given temperature, in addition to high mean effective pressures, with moderate size of pump.

When the initial temperature is high, there are two additional cases.

(2) The retention of a high mean effective pressure and a small pump with the maintenance of high efficiency demand that the initial pressure of the mixture, p_1, remain high and that the supply of air, N, be kept at a minimum. This case is equivalent to the preceding one except that the temperature, t_1, is raised.

(3) If the highest efficiency is required, at the expense of an increased bulkiness of the engine, the initial pressure of the mixture, p_1, may be reduced and the supply of air, N, increased.

A scientific investigation to determine the practicability of utilizing the products of combustion mixed with steam was undertaken by C. M. Garland in the Mechanical Engineering Laboratory of the University of Illinois from 1905 to 1907. The "constant pressure generator" devised for this investigation used liquid fuel, which of course gave no trouble from ash. Difficulties arose, however, to make the tests incomplete. A preliminary report of this work appeared in The Engineering Magazine for August, 1906. The results of the tests and an analysis of the investigation are on file in the laboratory, but they were not considered sufficiently complete for publication.

The plan of providing a mixture by forcing high temperature air into the boiler from coils of pipe in the hot flue was given a great deal of publicity about 1870.* There seems to be some contention as to the originator of this plan, but George Warsop of Nottingham, England, was active in promoting it as his and it became generally known as "Warsop's System." The plan was brought to the attention of the British Association in 1869 by Richard Eaton, who presented a paper on the subject a few months later to the Institution of Mechanical Engineers at Birmingham. Economy tests were reported, all of which indicated wonderful results in favor of the "system" as suggested by the following data:

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* Eng. (London), v. 7, p. 408, June 18, 1869: v. 8. p. 122, Aug. 20, 1869; p. 144, Aug. 27, 1869: v. 9, p. 78, Feb. 4, 1870; p. 260, April 15, 1870: v. 14, p. 92, Aug. 9, 1872: v. 16, p. 283, Oct. 3, 1873.

The Engr., (London)), v. 34, p. 348, Nov. 22; p. 381, Dec. 6; p. 431, Dec. 27; 1872: v. 36, p. 222, Oct. 3; p. 258, Oct. 17; p. 273, Oct. 24; 1873.

Van Nostrand's Eclectic Eng. Mag., v. 2, Aug., 1869; Dec., 1869: v. 9, p. 547, Dec., 1873.

Proc. Inst. M. E., 1870, p. 229.

	Air and Steam combined	Steam only
Even pressure trial		
Coal consumed	112 lb.	112 lb.
Weight of brake	120 lb.	120 lb.
Duration of experiment	153 min.	112 min.
Rev. of brake	15433 total	10500 total
R.p.m. of brake	101	93.75
Gross h.p. of useful work	972.55	661.79
Water evaporated	93.75 gal.	68.75 gal.
Weight of fire left in furnace	53.5 lb.	34 lb.
Open valve trial		
Coal consumed	140 lb.	140 lb.
Weight of brake	120 lb.	120 lb.
Duration of experiment	234 min.	196 min.
Rev. of brake	22815 total	17825 total
R.p.m. of brake	97.5	90.94
Gross h.p. of useful work	1428.05	1115.7
Water evaporated	131.25 gal.	112.5 gal.
Weight of fire left in furnace	43 lb.	28.25 lb.

The gain in power in the even pressure trial by using air and steam combined was 47 per cent, in the open valve trial 28 per cent. Tests in two different ships showed a saving of 27 per cent and 30 per cent respectively in fuel consumption because of the air injection. Later tests with stationary apparatus gave results as follows. The fuel consumption of a Nottingham boiler per horse power per hour with steam alone was 5.53 pounds while with air injection the fuel consumption was 3.40 pounds—a saving of 38 per cent. With a London boiler using steam alone the fuel consumption was 5.88 pounds, and with air injection it was 4.72 pounds—a saving of 20 per cent. Other brake tests showed gains of 24.5 per cent at a speed of 117 revolutions per minute, and 33.5 per cent at a higher speed.

Warsop's system was finally adopted by the Lancashire and Yorkshire Railway, and also installed successfully in several vessels. It was reported that 10 or 12 per cent of air in the mixture gave the best results. The principal advantages claimed for the system, apart from those which might result from mixing air and steam, were increased boiler efficiency from agitation and from better circulation of the water, and prevention of incrustation and priming.

The second plan of providing a mixture for air-steam engines, that of introducing the air after the steam has left the boiler, was described by William H. Shock, Chief Engineer, U. S. N., in a paper published in the Journal of the Franklin Institute in 1854.[*] The engine was invented by William Mount Storms of New York, and was commonly called "Storms' Cloud Engine."[†] Air compressed to approximately 18 pounds pressure was admitted to the cylinder at the beginning of the stroke for about 1.5 inches of piston travel; then steam at 80 pounds pressure was admitted during the next 2 inches of piston travel, when communication to the cylinder was closed and the mixture expanded. The test reported shows an increase in power due to mixing of more than 50 per cent. In the light of present knowledge, however, some of the statements made regarding the operation of the engine lead one to discredit this report.

Certain experiments by T. McDonough of New York, pertaining to the effect of mixing air with low-pressure steam, were reported in 1865 by the Journal of the Franklin Institute.[‡] Steam was generated in a small boiler and allowed to escape through a nozzle and to strike upon the fan of a small air mill. The quantity of steam used in a given time and the speed of rotation produced were noted; then air under moderate pressure was allowed to flow from a gas holder into the rear of the jet and to issue with the steam. From the amount of air added and the rotative speed of the mill it was found that the addition of about one part of air to ten parts of steam would increase the number of revolutions of the mill by one-fourth. It was suggested as an explanation that "the introduction of cold air might condense some of the steam, so causing it to be projected in liquid drops against the wheel, by which means its motive power would be more efficiently applied than when it issued as an expanding and diffusable vapor."

At about the same time facts similar to those just given were observed in connection with the operation of a small steam turbine,[§] by R. E. Rogers of the University of Pennsylvania.

Early in 1867 there appeared an announcement of Parker's steam and air engine.[¶] In this device heated air was injected or drawn into the steam pipe by specially designed nozzles and receiving cones; consequently the mixture reached the engine at a reduced pressure, about one-third of the boiler pressure. The tests reported did not include data on fuel consumption; hence no really convincing test results were presented; furthermore, it is difficult to conceive any increase in efficiency or economy by using such a method.[*]

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[*] "Notes on the Cloud Combination of Steam and Air," Jour. Franklin Inst., v. 58, p. 342, 1854.

[†] Jour. Franklin Inst., v. 60, p. 81, 1855; Eng. (London), v. 8, p. 258, Oct. 15, 1869.

[‡] Jour. Franklin Inst., v. 80, p. 282, 1865.

[§] London Mech. Mag., p. 7, Jan., 1865.

[¶] Eng. (London), v. 3, p. 61, Jan. 18, 1867.

Nineteen years later there appeared in the "Engineer"[†] a statement concerning experiments on Parker's engine which extended over a period of two days. During the first day tests were made with steam used as the medium and on the second day with air injected into the steam pipe. The consumption of fuel per horse power hour was practically the same for all tests. A detailed report of these tests could not be found.

In 1885 experiments were made by Zeller and Hurd[‡] at the New York Navy Yard for the purpose of ascertaining how much economic gain, if any, resulted from the use of a mixture of saturated steam and of compressed air in a non-condensing engine, the air being at the pressure of the steam and containing the heat of compression. The air cylinder and the steam cylinder were in tandem. Compression of the air started when the steam cylinder began to take steam; consequently the air pressure was at its minimum when admission to the steam cylinder took place. Thus no air could enter the steam cylinder until the pressure of the air became greater than that of the steam when further increase in air pressure caused the exclusion of the steam. It is reasonable to conclude, therefore, that this engine was not operated by a *mixture* of steam and compressed air, but rather by steam and compressed air used *successively* and *separately*. Because of this fact the data secured from these experiments, which show practically no difference in power produced by using steam alone and by using steam and air combined, are not conclusive for intimately mixed air and steam.

Andrew Jamieson[§] at a meeting of the Institution of Engineers and Shipbuilders held in Scotland, on April 30, 1895, read a paper entitled "A New Departure in Steam Engine Economy, with a Description and Tests of Field's Combined Steam and Hot Air Engine." The tests of this engine showed a saving of 40 per cent. A hot air pipe was connected to the jacket and to each end of a single cylinder non-condensing engine. The air was forced through a series of heating pipes in the main flue by means of a Roots blower which was driven by the engine. Admission of air took place automatically as soon as the exhaust steam began to escape, and continued until compression commenced late in the stroke. This hot air arrangement undoubtedly diminished or prevented cylinder condensation; the results show a remarkably low steam consumption, 18.6 pounds per horse power per hour for an engine of the type used. They also show only slightly higher consumption, 21.4 pounds, at low load. The results expressed in terms of brake horse power, 24.26 at normal load, and 37.9 at quarter load, are not so striking probably because of the inefficiency of the engine and of the blower.

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[*] The Engr. (London), v. 29, p. 281, May 6; v. 30, p. 249, Oct. 7, 1870. v. 31, p. 1, Jan. 6; p. 48, Jan. 20; p. 59, Jan. 27; 1871.

Van Nostrand's Eclectic Eng. Mag., v. 3, p. 185 July, 1870.

[†] The Engr. (London), v. 61, p. 506, June 25, 1886.

[‡] Jour. Franklin Inst., v. 121, p. 401, June, 1886. Zeitschrift des Vereines deutscher Ingenieure, v. 31, p. 284, April 2, 1887.

[§] Trans., Inst. of Eng. and Shipbuilders in Scotland, v. 38, p. 291, 1894-5.

A recent attempt to produce an air-steam engine was made by E. J. Christy of Waterloo, Iowa.* The engine, rated at 300 horse power, was constructed by the Vilter Manufacturing Company of Milwaukee, Wisconsin. A five-hour economy test was reported, in which the steam used per hour was 26.35 pounds per horse power at an average of 66.6 horse power. Christy used a scheme of operating the engine on a four-stroke cycle principle, the first two strokes before admitting steam being occupied with taking in and compressing a charge of cold air. The heat of compression of the air produced a high temperature within the cylinder and thereby prevented initial condensation of the steam. It is very evident that the air itself performs no useful work; therefore it is doubtful if any gain in economy resulting from mixing would more than compensate for the increased power loss of the two additional idle strokes. Furthermore, an engine operated on the principle of the Christy Engine would necessarily have to be much larger for a given power and speed than one using steam alone, not only because of the two idle strokes, but also because of the smaller weight of steam that can be used when the clearance space is full of air.

An engine working on the same principle as the Christy Engine was patented by G. Schimming† in Germany several years prior to the announcement by Christy. The patent was acquired by the Actien-Gesellschaft für Eisengiesserei und Maschinenfabrication, formerly J. T. Freund & Co. of Charlottenburg, and an engine was built and tested in 1886 or 1887. No data of tests made on this engine are available, but it is stated* that it proved a failure because of the enormous steam consumption and the large amount of water exhausted with the steam and air mixture.

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* "The Christy Air-Steam Engine," Power, v. 34, p. 382, Sept. 5; p. 528, Oct. 3; p. 603, Oct. 17; 1911.

† Zeitschrift des Vereines deutscher Ingenieure, v. 30, p. 964, Oct. 30, 1886.

An interesting but indirect application of the use of air and steam together is found in a hoisting engine for handling the skip in a single shaft 3700 feet deep at the Franklin Junior Copper Mine at Boston, Michigan.† This engine was conceived by R. M. Edwards, General Manager of the Franklin Mining Company, and was built by the Nordberg Manufacturing Company. The air cylinders, 36 inches in diameter, are in tandem with the steam cylinders, 46 inches in diameter, the stroke being 72 inches. The valve controls are arranged to make the air cylinders inoperative during the hoisting period and to make the steam cylinders ineffective during the return of the skip. The degree of compression in the air cylinders regulated by the operator controls the speed of the descending skip. There are three large receivers in which all the air compressed by the machine is mixed with the steam from the boilers. The mixture is then used in the steam cylinders during the hoisting period. A reducing valve in the steam supply pipe maintains about 65 pounds pressure on the receivers. During the fall of the skip the pressure in the receivers increases to approximately 95 pounds by the addition of compressed air, which is sufficient to lift the hoist about 1000 feet. Steam alone, at a pressure of 65 pounds, then completes the hoisting period.

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* Power, v. 34, p. 603, Oct. 17, 1911.

† "Air-Balanced Hoisting Engine," Power, v. 44, p. 640, Nov. 7, 1916.

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ENGINEERING EXPERIMENT STATION

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A STUDY OF COAL MINE HAULAGE IN ILLINOIS

BY

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A STUDY OF COAL MINE HAULAGE IN ILLINOIS

I. Introduction

1. *Scope of Present Discussion.*—Very few, even of those connected with the coal mining industry, appreciate fully the importance and extent of the underground haulage problems in a modern coal mine. The transition from mule haulage to modern electric locomotive haulage has been so rapid that there has not been time for most of those engaged in operating mines to study haulage practice in detail as has been done in connection with surface railroads; for example, in the tabulation of the ton-miles performance per locomotive per day, and similar statistical information. When one considers that at a large mine in Illinois 6000 or more tons of coal per day are hoisted in 5-ton capacity cars and that 1200 or more cars per day, or 150 per hour, must therefore be gathered from different parts of the mine, concentrated at the shaft bottom, loaded upon the cage over only two tracks, hoisted to the surface, lowered to the shaft bottom, and again distributed to remote parts of the mine, one realizes that here is a condition demanding thought and study if the most effective operation is to be secured from expensive equipment.

The coal mines of Illinois afford an unusually favorable opportunity for a study of the haulage problem, for not only are they the largest in point of output of any coal mines in the world, but there are few if any other coal fields of equal size where the operating conditions are so uniform. Beginning with primitive methods and equipment, the coal industry in the state has grown steadily until Illinois ranks third in coal production in the United States. The owners of the mines have not only kept pace with those of other regions, but they have in many instances been pioneers in installing improved equipment such as car lifts, self-dumping cages, and improvements about the shaft bottom.

An effort has been made in the present bulletin to trace briefly the development and history of mine haulage in general and in Illinois in particular. Mine haulage practice and costs have been considered under the three natural phases of shaft-bottom activities, main-line haulage, and gathering; while particular attention is called to Tables 4, 5, 7, and 8 which give the results of detailed studies in a number of the large producing mines of the state, that is, those with 3000 to 6000 tons per day output. These mines were studied in considerable detail and the results as presented in tables and graphs show that there is a wide diversity in the results obtained in mines of like capacity, with similar equipment, and operating under similar natural conditions. The tables suggest that a more detailed study of operating conditions in a number of mines of the state would produce a greater efficiency in operation even with the equipment already installed. This applies not only to the mechanical results obtained, as measured by performance in ton-miles etc., but also to the variations in costs for mines similarly equipped.

Approximately one-seventh of all coal mining employees are engaged in underground haulage duties, classified under 46 different occupations on the account books of different companies. In the present discussion of the subject, however, haulage is assumed to stop when the car is placed on the cage to be hoisted, thus excluding hoisting, although in the matter of cost it is not always possible to separate the hoisting cost from the haulage. In such cases, however, the hoisting cost is relatively small and does not materially affect the total haulage cost. Owing to the diverse accounting systems employed by different companies it is difficult to obtain comparative data for different mines, although the owners of the mines and the local superintendents have been most obliging in extending privileges for investigating haulage operations and in supplying information relative to operating costs.

Every study of an industrial problem should include a consideration of the accidents connected with the industry; therefore some discussion of accidents in mine haulage, based upon the statistics given in the Coal Reports of the Illinois Department of Mines and Minerals, is included in this bulletin. An analysis of these statistics has been made in the effort to show the relation between coal production, number of employees, and the number of fatalities due to haulage operations among various classes of mine employees.

2. *Acknowledgments.*—This bulletin is the outgrowth of a study of Mine Haulage undertaken as a research problem under the direction of the Engineering Experiment Station of the University of Illinois by A. C. CALLEN* while Associate in Mining Engineering at the

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* Now Professor of Mining Engineering, University of West Virginia.

University. He prepared much of the historical material and some of the statistics for accidents that occurred prior to 1917.

Upon the resignation of Mr. Callen the study was continued under the Coöperative Coal Mining Agreement between the Engineering Experiment Station, University of Illinois, the United States Bureau of Mines, and the Illinois State Geological Survey.

The field studies of haulage operation were carried on mainly by J. R. FLEMING, who, together with A. J. HOSKIN, prepared the tables and graphs giving the results of these field studies. Mr. Fleming also supplemented the studies of accidents made by Mr. Callen. Final arrangement, checking, and editing of the manuscript was done by A. J. HOSKIN and H. H. STOEK.

The authors gratefully acknowledge the hearty coöperation of the owners and operating officials of many of the mines in the state in giving assistance, not only through replies to requests for information by mail, but also in carrying on the studies in the mines and in permitting free access to the books of the companies in order to obtain costs of operation. They are also indebted to J. J. RUTLEDGE, Superintendent of the Urbana Station of the United States Bureau of Mines, and F. W. DEWOLF, Chief of the Illinois Geological Survey, for suggestions during the progress of the investigation, and for their careful review of the manuscript.

II. Evolution of Mine Haulage

3. *Early Practices.*—The primitive method of transporting material from underground mine workings was for men to carry it in some form of container, as a tray.* Similar methods are still used in a few places where the natural conditions of the mineral deposit make them necessary, or where they are economically possible.† The introduction of wooden sleds was an improvement over carrying. Such sleds, or baskets, provided with runners and usually drawn by boys, were extensively used in Great Britain in early coal mining, and are still used in thin seams where the expense of taking down the roof to obtain necessary head-room for cars is prohibitive.‡

The introduction of wheeled vehicles was the next advance step. By using a wheelbarrow heavier loads could be moved with much less exertion than by carrying, especially if a plank road was used instead of the natural mine floor. Although wheelbarrows are still used in many ore mines, they are seldom found in coal mines.

The four-wheeled truck or car soon replaced the wheelbarrow for general use. At first, wicker baskets or wooden tubs were loaded at the face and carried to the haulage road, but soon cars or ''waggons'' were made of such a size that they could be taken to the face. The ''buggies,'' still used in Kansas longwall mining for transporting the coal from the advancing face to the road-head where it is transferred to the regular mine cars, are illustrated in Fig. 1.§ This buggy is run along the longwall face on eight-pound steel rails. The track is made up in eight-foot sections with a curve section for the road-head, so that it can be easily handled.

In England the term ''tub'' is still used for a mine car though very few real tubs are used. Pushing cars by hand is known as ''putting'' in England and as ''tramming'' or ''hand tramming'' in the United States.

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* Agricola, "De Re Metallica." Book VI, p. 56, translated by H. Hoover.
A. Pliny (XXXIII, 21).

† Tonge, J. "Principles and Practice of Coal Mining," p. 162, London, 1906.

‡ Hughes, H. W. "A Text Book of Coal Mining," p. 224, London, 1904.

§ This photograph is furnished through the courtesy of C. N. Fish, general manager of the Home Riverside Coal Mines Co., Leavenworth, Kansas.

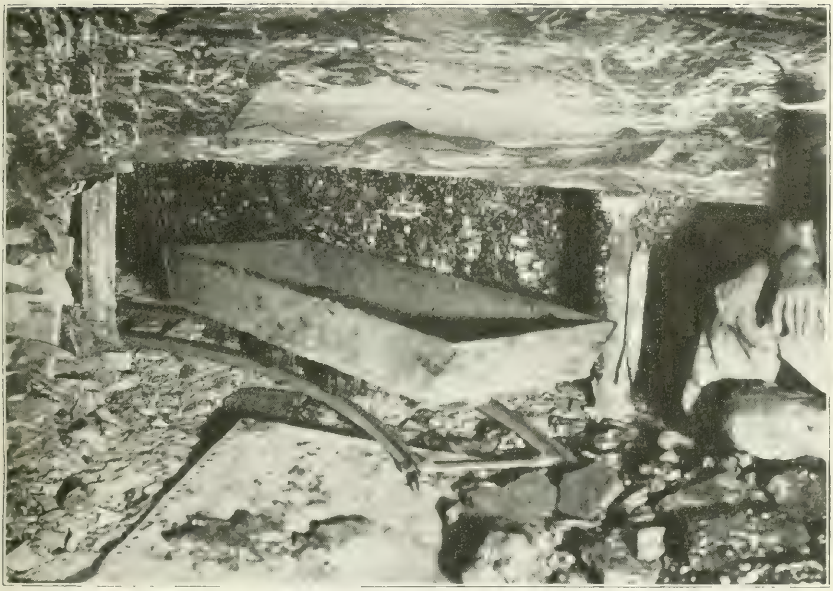

FIG. 1. WHEEL BUGGY IN KANSAS COAL MINE

"Cast-iron tram plates were introduced in English mines in 1767 and were in turn succeeded by wrought iron rails and steel rails."*

The modern coal-mine car bears little resemblance to the early "tubs." Samuel Dean, an English mining engineer who has written extensively on the coal mines of the United States, attributes the larger output per man in the United States to the larger capacity of cars used.†

4. *Hand Tramming.*—Hand tramming, by which is meant the manual pushing of cars or trucks, was among the earliest systems of transporting mined material. At present it is used mainly in coal mines of small capacity where the working face is not far from the shaft bottom or drift opening, or in places where the height of entry is too small for animal or mechanical haulage. In some coal mines miners push the empty cars from a distributing parting to the working face; in others, though less frequently, loaded cars are pushed from the face to the parting where they are formed into trips for animal or mechanical haulage. This system requires suitable grades and cars of such capacity that they can be moved readily and easily kept under control. The amount of hand tramming in unionized mines is generally stipulated in the agreement between the miner and the operator.

5. *Animal Haulage.*—Following the enactment of a law prohibiting the employment of women or of children under 10 years of age, Shetland ponies were introduced in English mines in the year 1843, as substitutes for the putters employed in conveying the coal from the working face to the main roads.‡ Where coal seams were thicker, horses were employed and in England they are still the favorite animals for underground labor. In the United States mules are generally preferred to horses as they are quicker and more surefooted. Dogs have been used in small Illinois mines for hauling coal. In one Ohio mine they are said to have been used for over thirty years. Overman, in his "Metallurgy" published in 1852, says that the dog-cart was at that time in general use in coal mines of the western United States and was a most convenient vehicle for

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* Foster & Cox, "Ore and Stone Mining," p. 373.

† Trans. Inst. Min. E. Vol. 50, p. 179.

‡ R. L. Galloway, "Annals of Coal Mining." Vol. 2, p. 344.

handling coal underground.* Oxen have been used for coal haulage to a very limited extent.

Mules, horses, and ponies are still widely used in the mines of the United States. Although sometimes employed in main haulage their chief use is in gathering cars on short hauls, that is, in taking the loaded cars from the working face to a parting where the cars are made into trips for transportation by mechanical means.

6. *Mechanical Haulage.*—The principal forms of mechanical haulage now in use are rope haulage and locomotive haulage.

Rope Haulage

Rope Haulage may be divided into four systems: Engine Planes, Gravity Planes, Endless Rope, and Main and Tail Rope.

An Engine Plane is an inclined plane up which a load is drawn by an engine or motor. Such a plane may work ''in balance,'' the empty cars descending while the loads are coming up, thus partially balancing the system and reducing the load on the engine; or the system may work ''unbalanced,'' in which case the engine simply draws the loaded cars up the plane while the empty cars pull the rope down again.

The earliest adoption of mechanical haulage underground was about 1812 or 1813, when George Stephenson so altered an underground engine at Killingworth colliery, England, as to make it haul the coal up an inclined plane to the shaft.† Chains were originally used to haul the cars up. About the year 1841 ''the haulage of coal by ropes was greatly facilitated by the introduction of light, round iron wire ropes.''‡

A Gravity Plane is one of such inclination that the loaded cars going down the plane pull the empty cars up, the inclination being usually over 20 per cent. Such planes are used where the coal must be transferred to a lower level; for example, in mountainous regions where the mine openings are located at elevations above the tipple, and inside mines where the coal beds are steeply inclined.

An Endless Rope haulage system has an endless rope that is operated continuously by a haulage engine at a speed of usually two to four miles per hour. The cars are attached to the rope, either singly or in groups, by grips or clamps which can be easily fastened to or unfastened from the rope. Two tracks are required, one for the loads and the other for the empties. The system is used mainly on short hauls and on steep pitches such as slope openings.

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* Col. Eng. Vol. 20, p. 684.

† Galloway, R. L. "Annals of Coal Mining." Vol. 2, p. 340.

‡ Galloway, R. L "Annals of Coal Mining." Vol. 2, p. 344.

In the Main and Tail Rope system a trip of loaded cars is pulled by the main rope, a tail rope being fastened to the rear end of the trip and dragged after it. At the destination the ropes are uncoupled from the cars and the tail rope is fastened to the front end of the empty trip while the main rope is fastened to the rear end. The trip of empty cars is then pulled in by the tail rope and the main rope dragged after the trip. The speed of operation is usually 6 to 10 miles per hour. The system is used mainly on a haulage road having undulating grades.

Rope haulage is said to have been introduced in the United States about 1870.*

Locomotive Haulage

The different types of mine locomotives that have been used are steam locomotives, compressed-air locomotives, gasoline locomotives, and electrical locomotives.

The exact date of the first use of steam locomotives in connection with underground mining in the United States is not definitely known, but according to E. B. Wilson it was prior to 1870.†

On account of the smoke and other products of combustion, such locomotives should be restricted in their use to the return air-ways. At one time they were extensively used in the anthracite region of Pennsylvania and from 1883 to 1895 in the Pocahontas region of West Virginia, but they have never been used in Illinois. There are still a few steam locomotives used in the Pocahontas district at three or four of the small mines where the tonnage remaining to be mined does not warrant the expense of a change to electric haulage.‡

From 1875 to 1895 may be called the experimental period of the compressed-air locomotive. Ten or twelve were built during this twenty-year period and were installed by operators who desired a haulage system that would eliminate fire risk, be free from the dangers of electric wires, and be comparatively safe in a gassy mine. In construction compressed-air locomotives differ from steam engines mainly in having, instead of a steam boiler, a large storage tank which can be charged with air at a pressure of from 600 to 1000 pounds per square inch, and a reducing valve set to supply air to the cylinders at a constant pressure of 150 pounds. From 1895 to 1908 great improvements in design and manufacture were made, and several hundred locomotives were furnished to mining companies.

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* Mines and Minerals. Vol. 31, p. 71.

† Mines and Minerals. Vol. 31, p. 71.

‡ Private communication, Lincoln, J. J.

In 1908 the first two-stage compressed-air locomotive was put upon the market and, in the three years succeeding, over 100 were built.*

According to the H. K. Porter Company of Pittsburgh, Pennsylvania, there were in 1921 no compressed-air locomotives operating in the coal mines of the Mississippi Valley, but in the mines of Western Pennsylvania and West Virginia the total number was about 150.

There are two great advantages of compressed-air locomotives: first, they are comparatively safe for use in gassy mines, and second, they require neither trolley wire nor rail bonding.

On the other hand they are bulky, and their radius of operation is limited by their air-storage capacity. However, in mines having ample cross-section of the entries this is not serious as tanks of a capacity sufficient for a run of several miles may be used.

The advantages of a locomotive carrying its own source of power, such as a gasoline locomotive, are obvious. It was but natural that an attempt should be made to use the internal-combustion engine for mining service and, indeed, before the automobile had advanced much beyond the experimental stage, a gasoline locomotive was tried out for hauling coal.

Probably the first gasoline mining locomotive made in this country was furnished in 1898 by W. F. Prouty of Philadelphia, Pennsylvania, and Newark, New Jersey, to the St. Bernard Mining Co. for use in the No. 9 mine at Earlington, Kentucky.† This locomotive was in service for a year, but was never able to pull a full trip of loaded cars and was finally scrapped.

It is likely that gasoline locomotives had been in use in Europe for some years previous to this date. In 1899, in describing the explosion-proof gasoline motors used in the coal mines of Belgium,

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* Mines and Minerals. Vol. 31, p. 365.

† Coal Age. Vol. 5. p 9.

M. J. Kersten said, "It is only quite lately that a locomotive working with petroleum has been used in fiery mines,"* the presumption being that they had been used for several years in non-gassy mines.

The first gasoline locomotive used in Illinois was probably the second successful one in this country. It was built by the Sangamon Coal Co. and put in its mine at Springfield, 1904. This crude machine pulled a trip of seven to nine mine cars, each weighing, when loaded, 4000 pounds. A few locomotives of this type were built in Chicago and in St. Louis about 1905 or 1906, but the St. Louis locomotives were returned to the manufacturers as they cost more for repairs than the value of the coal they hauled. A few gasoline mine locomotives were made by Fairbanks, Morse & Co. in 1907.†

In 1909 gasoline locomotives were introduced into the lead mines of southeastern Missouri where the Desloges Consolidated Co., on account of its very excellent ventilation, was able to use them with success.‡ The George D. Whitcomb Co. shipped one to the Kolb Coal Co. of Mascoutah, Illinois, in 1909. This locomotive gave such satisfaction that several more were ordered by this company. In 1910 it was stated§ that there were three hundred of these locomotives in use in all parts of the world. In 1915 about that number were in use in the United States.

Although gasoline locomotives have the great advantage of flexibility and cheapness of installation, their use underground has been restricted because of the possible danger from the exhaust gases and from the extra mechanical attention necessary to keep them in operating condition. Their use underground is steadily decreasing; storage-battery locomotives are replacing them to a very great extent. Gasoline locomotives are restricted in their use to main-line haulage and in this to return air-ways only.

In 1914 the United States Bureau of Mines conducted an investigation into the vitiation of mine air resulting from the use of gasoline engines. According to the conclusions of the Bureau,¶ the ventilating current—in order to safely dilute the obnoxious carbon monoxide exhausted from a gasoline locomotive—should be increased to the extent of from 2610 to 35 140 cubic feet per minute, this additional volume of air depending upon the size of the engine and the thoroughness of the carburation. These figures are based upon the dilution of the poisonous gas to one part in 1000 parts of air, this quality of atmosphere being safe for men and animals to breathe for "short and infrequent intervals" only. For continued conditions the dilution should be to not more than one part of the engine's exhaust in 2000 parts of fresh air. It will be seen that this feature of gasoline locomotives is a serious objection to their use underground even upon return air-ways. In the attempt to restrict the pollution of the mine air experiments have been made with passing the engine exhaust through chemical solutions but the results were unsatisfactory.

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* Eng. and Min. Jour. Vol. 68, p. 724.

† Illinois Coal Mining Investigations. Bul. 13, p. 179.

‡ Eng. and Min. Jour. Vol. 84, p. 346.

§ Mines and Minerals. Vol. 31, p 30.

¶ Hood and Kudlich, U. S. Bureau of Mines. Bul. 74, Gasoline Mine Locomotives in Relation to Safety and Health, p. 7.

The first electric locomotive using current from a dynamo was built by Siemens and Halske in Germany, and, at the Berlin Trade Exhibition in 1879,* was operated upon a circular track about 1500 feet long. The introduction of electric locomotives into mining service followed almost immediately, and in 1882 the first electric mine locomotive was installed in the royal coal mines at Zaukerode, Saxony.† This system of haulage was adopted by the Consolidated Paulus and Hohenzollern Collieries at Beuthen in 1883, and at New Stassfurt in 1884. The locomotives were all built by the Siemens and Halske Co. On July 26, 1887, the Lykens Valley Coal Co. put the first electric mining locomotive in this country into service at the Short Mountain Colliery at Lykens, Pennsylvania.‡ This locomotive had a 30-horsepower motor wound for 400 volts direct current. The conductor was a 25-pound iron rail mounted along one side of the entry, current being taken off through four contact wheels. The motor and running gear weighed 1500 pounds, but the machine was weighted with scrap iron up to 7000 pounds. (See Fig. 2.) It was built by the Union Electric Co. of Philadelphia, Pennsylvania. This installation was the first of any considerable size in the world. The Siemens and Halske locomotives weighed only two tons each and hauled a train load of about 10 tons, while the Lykens Valley "Pioneer" hauled a load of 150 tons at a speed of six miles per hour over a road about 6300 feet long. It was still in service in 1915. In

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* Sprague, F. J., Elect. Ry., p. 3. Int. Eng. Cong. 1904, p. 3.

† Electric Locomotives in German Mines. Karl Eilers, Trans. A. I. M. E. Vol. 20, p. 356.

‡ Col. Engr. Vol. 8, p. 43. Also Thesis of H. H. Stoek.

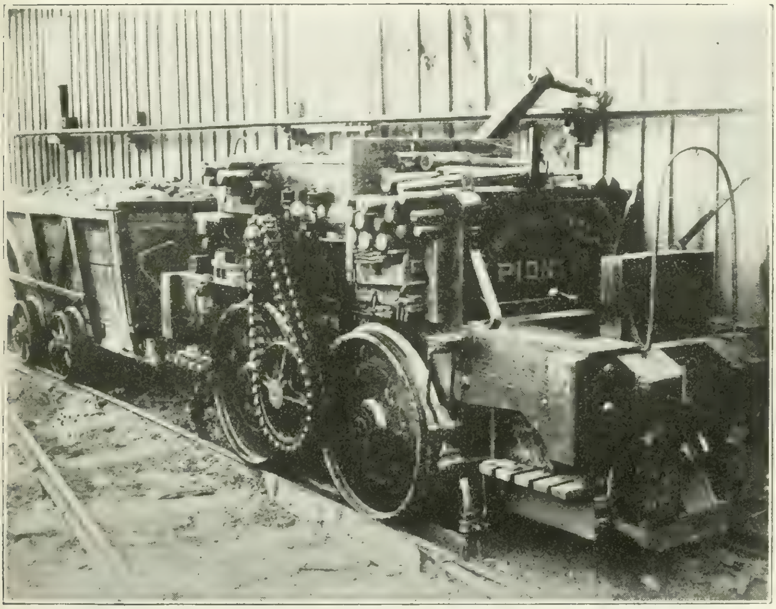

FIG. 2. FIRST ELECTRIC MINE LOCOMOTIVE IN UNITED STATES

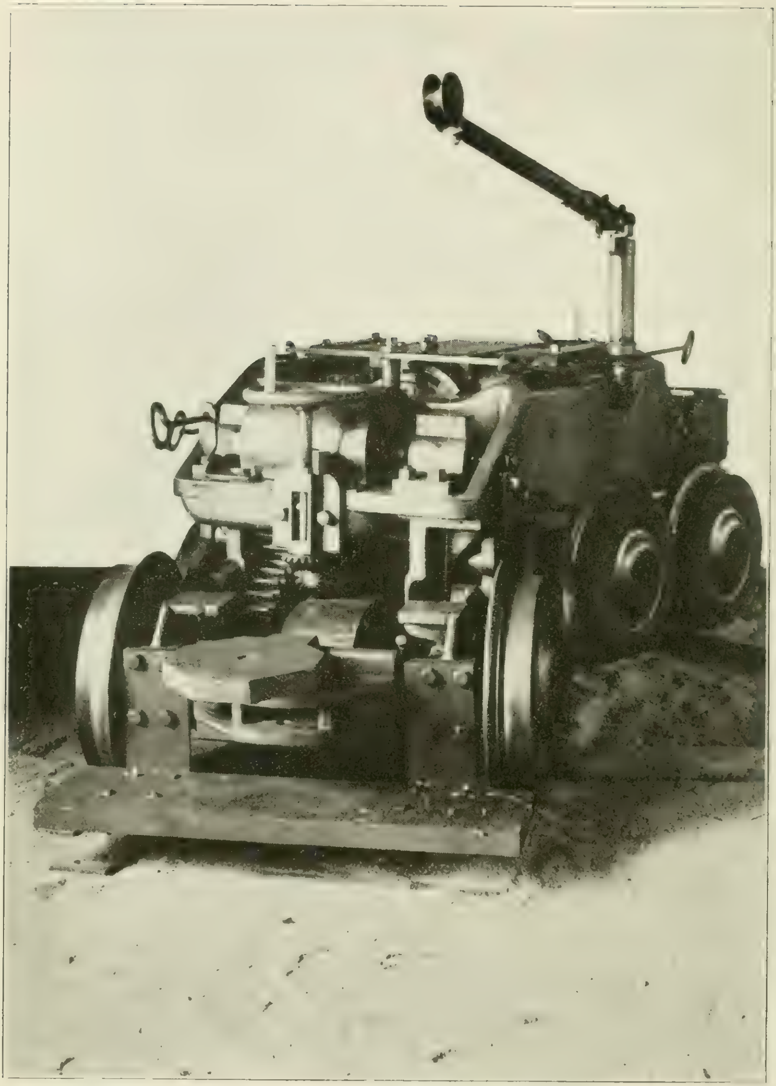

Fig. 3. First Electric Locomotive in Illinois Mines

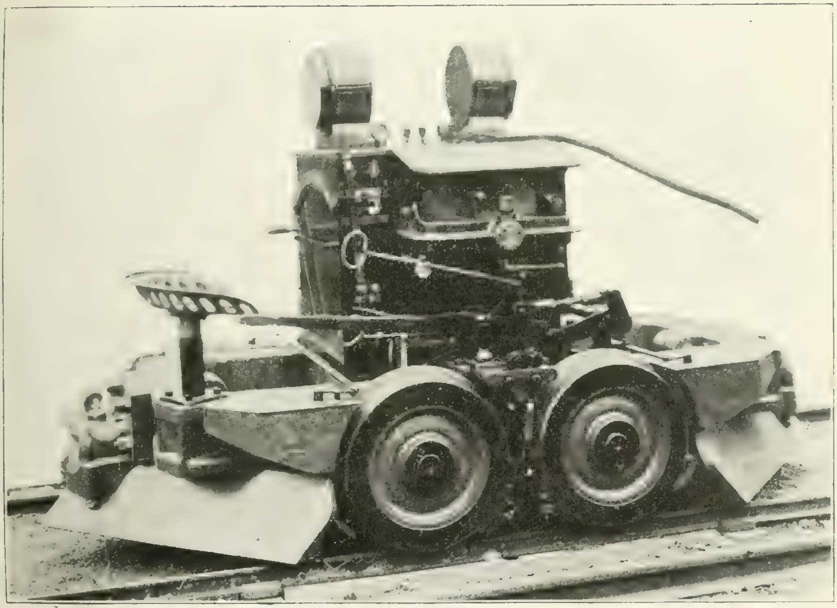

FIG. 4. EARLY TYPE LOCOMOTIVE USED AT CENTRALIA, ILLINOIS, IN 1899

1888 the Jeffrey Manufacturing Co. built the first electric locomotive used in a bituminous coal mine in the United States. This installation was in the mines of the Upson Coal Mining Co., Shawnee, Ohio. Instead of a wire or rail as a conductor two parallel 1-inch galvanized iron pipes were used. The rails were not bonded, as one of the pipes was used for the return circuit.

The first electric mine locomotive installed in the State of Illinois was placed in the No. 3 mine of the Chicago, Wilmington & Vermilion Coal Company at Streator in 1888. This locomotive, Fig. 3, was designed by Elmer A. Sperry of the Sperry Electric Mining Machine Company of Chicago, (the predecessor of the Goodman Manufacturing Company), and was built by that company. This was an experimental machine and was provided with eight driving wheels and a motor of about 30 horsepower. The total weight of the machine was about six tons. Referring to this first electric mine locomotive in Illinois, C. A. Pratt, Chief Engineer of the Goodman Manufacturing Company, says:* "It was in operation several months and was then replaced by a locomotive of somewhat modified design and of greater weight and horsepower. The locomotive which replaced it had eight driving wheels distributed on two bogey trucks. These wheels were about 20 inches in diameter and the locomotive was designed to turn on a curve of 8 or 9 feet radius. The locomotive weighed about 8 tons and was driven by one 60-horsepower motor, the armature of which was geared to all of the eight wheels. A second locomotive of the same description was put into the same mine some months later and these two locomotives were operated for many years." As far as can be learned this installation at Streator was the only really successful one for several years, though some locomotives had been used experimentally at other mines. No further introduction of electric haulage was made in Illinois until 1899 when the Jeffrey Manufacturing Co. shipped an 8-ton locomotive, Fig. 4, to the Centralia Mining and Manufacturing Company of Centralia, Illinois.

The years 1899 to 1904 may be called the introductory period. The increase in installations was slow but steady so that by the close of this period each of the important mining districts in the state had at least one mine in which electric locomotives were being used with success.

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* Private communication.

The introduction of the electric locomotive and its successful operation in main haulage led to attempts to extend this system to gathering service. In early practice miners pushed their loaded cars to the room necks whence the cars could be hauled to main partings by trolley locomotives. When rooms were driven to the rise this practice occasionally involved accidents from runaway cars. A locomotive was therefore needed to do such gathering safely, but of a type that required no trolley extensions into the rooms. In response to this need the cable locomotive was designed. Briefly defined, this locomotive is one that can not only operate as a trolley locomotive but also travel on track not equipped with trolley wire by taking its power through a long flexible conductor or cable that it carries mounted on a drum or reel.

Probably the first successful cable locomotive was constructed in 1900 in the shops of the Pocahontas Consolidated Collieries Co. at Pocahontas, Virginia.* For several years previous this company had been trying to develop a storage-battery locomotive but without success. So, in 1900, they mounted on one of these old locomotive frames, with the motor, a vertical cable reel, thus making a very good gathering locomotive known later as the ''Wampus'' on account of its peculiar appearance. Since 1900 all electric locomotive manufacturers have constructed gathering locomotives, the designs being generally similar. The cable through which the locomotive receives its power when away from the trolley wire is wound either on a reel placed horizontally on top of the locomotive, or on a drum placed at one end. The reel or drum is driven by an independent motor, by a spring device, or by a chain and sprocket wheels from the axle.

Where the rooms dip rather steeply towards the face it may be impossible or undesirable for the locomotive to go to the face for the cars. In such instances the ''crab'' locomotive has been used with success. This locomotive is equipped with a drum on which a steel cable is wound and which is usually driven by a separate motor, thus in reality adding to the locomotive a small hoisting engine for the purpose of pulling cars out of steeply pitching places while the locomotive remains on the entry. Under some conditions a gathering locomotive is equipped with this ''crab'' device in addition to the cable attachment.

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* Mines and Minerals. Vol. 30, p. 13.

The rack-rail locomotive was devised for electric haulage on heavy grades. Its hauling capacity is not limited to the adhesion between the wheels and rails. Instead of driving the wheels the motor is geared to a sprocket wheel beneath the locomotive, the teeth meshing with a rack-rail laid between the main rails. The locomotive is therefore really geared to the track and can haul large loads on steep grades, provided the strength of the parts and the power of the motor are sufficient. Rack-rail locomotives were first brought out by the Morgan-Gardner Co. in 1899. They are used in mines where the grades are prohibitive to ordinary electric haulage. In some cases no trolley wire is used, the rack-rail acting as a conductor for the current. In other cases a trolley wire is used on the ordinary haulage roads, the rack-rail being used only on occasional grades.

From the time that electric haulage was first introduced in mines it has been the desire of engineers to find some way of dispensing with the trolley wire and the bonding of the rails; first, from a desire to save the outlay required by such an installation, and second, because of the danger from contact with the wire, and from explosions caused by sparking of trolleys and wheels in gassy mines. As regards the latter danger the United States Bureau of Mines at the Pittsburgh Testing Station is prepared to test locomotives in a gas chamber and, if they can comply with requirements, to list them as permissible for use in gaseous atmospheres. It is doubtful, however, if any trolley or reel locomotive can meet these requirements. This condition, therefore, led to the introduction of the storage-battery locomotive, which, while it does not eliminate the danger from switch and motor sparks, at least dispenses with the trolley wire.

The commercial development of the storage battery began at about the same time as did that of the electric railway, for it was not until 1880 that Brush and Faure, working independently, simultaneously produced the pasted plate for storage batteries, resulting in lighter and cheaper cells. Naturally the storage battery was looked to as the solution of the problem of dispensing with trolley wires or other naked conductors. The early development of such locomotives took place in England and in Germany, American engineers being slow to take up the subject. In 1886 the first storage-battery locomotive was tried in the mine of the Trafalgar Colliery Co.* Indicative of the slow development of the early installations was the statement made in 1895 that storage-battery locomotives had then reached the experimental stage only.* Probably the first successful use of a storage-battery locomotive in this country was at the mines of the Southwest Virginia Improvement Co. in the Pocahontas region of West Virginia.† The Baldwin-Westinghouse Co. built this locomotive in 1899 and it proved so successful that the company ordered six more. For several years prior to this the Pocahontas Consolidated Collieries Co. at Pocahontas, Virginia, had endeavored to develop a storage-battery haulage locomotive, three machines actually having been built which were, however, only "more or less effective."

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* Eng. and Min. Jour. Vol. 42, p. 98.

About 1900 the Jeffrey Manufacturing Co. shipped its first storage-battery locomotive. During the following ten years there were several locomotives of this type put into service but, on the whole, development was slow. Beginning with 1911 these locomotives began to attract a great deal of attention. Storage batteries had been improved both in design and construction. The Edison alkaline battery with a steel jar had been placed on the market and had given excellent service. Mining men did not need to be convinced of the advantages in the use of storage-battery locomotives, but they were extremely dubious about the ability of a battery to stand up under the severe conditions of mining service. In some storage-battery locomotives, particularly of the earlier types, batteries were too small and motors were of too low capacity for the weight of the locomotive. Whereas in main-haulage locomotives of the trolley type motors of approximately 10 to 12 horsepower per ton of weight are used, in some storage-battery locomotives the motor capacity has been as low as one horsepower per ton of weight. This radical difference restricts the continuous performance of the storage-battery locomotive for heavy work and necessitates extra care to maintain the batteries in proper working condition. This places storage-battery locomotives at a disadvantage as compared with trolley and cable-reel locomotives. Recently storage-battery locomotive manufacturers have shown a tendency to install larger motors than formerly.

These locomotives are fitted with motors that are built to withstand heavy overloads. Although their normal ratings may be relatively low they will stand without injury overloads of 300 to 400 per cent if not too long sustained. For instance, a certain four-ton machine is rated at 80 volts and 60 amperes when running at about 1050 revolutions per minute, this being equivalent to slightly more than six horse power.* However, it is not unusual for this machine on short hauls to consume 300 to 350 amperes. This practice is based upon the following general considerations: These light-weight storage-battery locomotives are used to do both gathering and main haulage. During the work of gathering the duty is light, perhaps 75 to 80 per cent of the working time being spent in hauling one car at a time to and from rooms. When a few cars have been collected in an entry they are hauled to a parting where a train is made up and the locomotive then hauls this train to the shaft bottom. Assuming that the average distance from parting to bottom is 2000 feet and that a speed of four miles per hour is maintained, the run will require less than six minutes. For such a short period these motors will easily withstand the overloads, which may be six or seven times the normal ratings.

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* Col. Engr. Vol. 16, p. 32.

† Mines and Minerals. Vol. 30, p. 13.

As regards electric mining locomotives in general, in earlier practice, when the hauls were short, seven and eight horsepower per ton of locomotive weight was commonly used; that is, from 70 to 80 horsepower on a 10-ton locomotive traveling at a speed of six miles per hour. As the requirements became more severe it was found that motors of such horsepower overheated, wherefore motor capacities were increased to a minimum of 10 horsepower per ton in general mining practice. As the loads to be hauled by the main-haulage locomotives increased, manufacturers increased the motor capacities to not less than 12 horsepower per ton for locomotives above eight tons rating. For long hauls it is now not uncommon to use still greater horsepower where the circumstances will permit. Under severe conditions mine locomotives may be required to develop in excess of 15 horsepower per ton, and such requirements are fulfilled successfully by applying forced ventilation to the motors.

The chief demand of mining men has been for increased locomotive capacity without increase in size. In discussing compactness of design G. M. Eaton, chief engineer of the Westinghouse Electric and Manufacturing Company, cites an electric mining locomotive built in 1896 that had a ratio of volume (cubic feet) to horsepower of 3.88, while a more modern locomotive of the same motor capacity has a ratio of 1.54.*

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* This is on the basis of 55-deg. temp. rise in 4 hr., and not on the A. I. E. E. restriction of 75 deg. in 1 hr.

Manufacturers have experimented to secure equal distribution of weight on the driving wheels; to prevent the slippage of one set of the wheels, when only one motor is used, by connecting the front and rear axles; to determine the best position of the drawbar to assure the most advantageous line of pull; to increase the effective drawbar-pull by increasing the weight of the locomotive; to so increase the number of driving wheels as to distribute the weight and reduce the load on each wheel; to make possible the use of tandem locomotives or of trailers upon which is carried all excessive weight (particularly that due to the use of the storage battery) so that the driving wheels will carry only the weight desired for the required pull; to introduce steel-tired wheels instead of cast-iron wheels in order to secure greater adhesion to the rail; to decrease the friction in the locomotive by the use of special bearings and improved methods of lubrication; and to mount independent motors in a storage-battery locomotive—one for trolley current, the other for battery current.

Improvements and changes in the design of electric locomotives have been made principally as follows:

(a) Details of construction, both electrical and mechanical, have been modified to better adapt the locomotives to severe mine service.

(b) Compactness has been sought to permit use of locomotives in restricted quarters.

(c) Increased capacity and endurance have been secured for the electrical equipment.

(d) Greater flexibility of movement has been obtained through the use of cables and storage batteries.

Among the modifications of details of mechanical construction may be mentioned the change from ordinary brass bearings to ball bearings for armatures; the use of heat-treated or hardened motor pinions; the making of all working parts much heavier to take care of the increased duties imposed upon them; and the making of such working parts more accessible and more readily detachable.

Locomotive frames were originally made of cast iron. These did very well unless collisions occurred, when repairs were difficult.

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* Development of Electric Mine Locomotive. Proc. A. I. E. E., April, 1914.

Later, cast steel came into use for frames and to a certain extent is used for parts of the frames by some builders today, although rolled-plate side-frames are more common. Rolled steel is more uniform than cast steel and it is less likely to contain blow-holes. Some builders have given special attention to the bracing of the frame at the corners to resist blows from collisions or derailment. One company equips its locomotives with an auxiliary buffer and interposes springs between it and the main locomotive frame to take up the shocks of collision, coupling, and starting. This construction results in a saving on car hitchings and bumpers and is of assistance in starting trips.

In the early locomotives axles were too weak, journal boxes were too short for the weight, journal springs were not sufficiently flexible to meet the conditions of mine track, motor suspensions were often too rigid to allow the wheels and axles to follow the track, and brake-riggings had springs that reduced the effectiveness of the brakes. Many early locomotives were made with a chain drive between the axles, but this method of driving has been abandoned by several manufacturers whose locomotives now have either a single motor geared to both axles, or two motors, one for each axle. One manufacturer, however, continues the chain drive, with good arguments for its superiority over direct gearing.

Amongst the improvements in electrical details may be noticed first the use of commutating poles on the motor to prevent sparking, and second, the thorough enclosing of the electrical parts, these changes at once reducing the danger of fire or explosion and increasing the life of the parts.

In many instances field-windings have been changed from cotton-covered wire to strap copper insulated between layers or turns with sheet asbestos and the whole wrapped with oiled linen, asbestos tape, or other fireproof insulation, baked with varnish. Formerly the fields would deteriorate from heating; now life is indefinitely prolonged. When necessary it is a comparatively simple matter to repair the strap coils without the loss of any copper, whereas, with the wire-wound fields repairs to defective or damaged insulation often required the purchase of new copper wire or new material throughout. The armature coils were generally wire frequently of two or more turns per coil, but today they are largely made of bar or strap copper of only one turn per coil. The repairing of this type of coil is very much simpler and the copper is usually salvaged, whereas with the old type a complete replacement of the damaged parts was generally necessary. The material used in insulation is of much better quality than that used heretofore, securing increased life of the coil. The single-turn coil results in better commutation and less sparking at the brushes than was possible with the older construction.

The improvement in locomotive controllers has been marked. Those now used are of the straight type without any auxiliary devices. The size and capacity of the blow-out coils have been greatly increased and, in the best designs, strap copper with fire-proof insulation is used.

On storage-battery locomotives it is considered best practice to have all switches in an enclosed compartment so that they can not be thrown when the locomotive is operated under gassy conditions. No attempt has been made to enclose the storage batteries themselves in explosion-proof cases, as circuits are not broken while the batteries are operating, and there must be ample ventilation about the batteries to carry away the gas generated therein.

An effort has been made to standardize practice in mine haulage through a committee of the American Mining Congress for the Standardization of Underground Transportation Equipment. Although the subjects that have been investigated by this committee—such as track gauge, minimum track curvature for rooms, wheel-base for mine cars, types of couplers, and overall dimensions of mine cars—apply primarily to track and mine-car construction, any standards adopted will affect locomotive design. The rating of mine-locomotive motors is generally governed by the rules of the American Institute of Electrical Engineers for railway-type motors. The rated horsepower delivered for one hour should not heat the windings more than 75 degrees C. above the surrounding air, Standardization Rule No. 415 being as follows:

> ''The nominal rating of a railway motor shall be the mechanical output at the car or locomotive axle, measured in kilowatts, which causes a rise of temperature above the surrounding air, by thermometer, not exceeding 90 degrees C. at any other normally accessible part after 1 hour continuous run at its rated voltage (and frequency in the case of an alternating-current motor) on a stand with the motor covers arranged to secure maximum ventilation without external blower. The rise in temperature as measured by resistance, shall not exceed 100 degrees C.''

The Electric Power Club has the following standard rule specifically applying to mine locomotives:

> "Mine locomotive motors shall be given nominal ratings which shall be the horsepower output at the armature shaft, excluding gear and other transmission losses, which the motor will develop for one hour under normal rated conditions on a stand test with covers removed and natural ventilation, without exceeding the temperature rises guaranteed."

In order that the motor shall have good continuous operating capacity, in proportion to its capacity on the hour rating, it is necessary to have a liberal radiating surface in addition to the usual requirements of ample area of conductors and commutator.

The manufacturers usually guarantee a certain starting drawbar-pull on clean, dry rails, and also running drawbar-pull at specified speeds.

7. *Mine Haulage in Illinois.*—Most of the large producing mines in Illinois are being operated in seams of coal which are usually over 5 feet in thickness, thus permitting the use of cars that are larger than the average used in the United States. The largest car in use holds about 5 tons and the average about 3 tons. With the exception of occasional heavy local grades the coal seams are nearly level. The floor is fire clay and affords a good road-bed. These conditions permit a systematic arrangement of haulage ways and favorable and efficient haulage. Because of these favorable natural conditions and because the more modern mines are all designed for large tonnages, large capital investments have been made, with the result that the more modern Illinois mines are exceptionally well equipped. Cars of the capacity noted above require a good track; therefore, in most of the mines developed during the past ten years 40-pound rails have been used on the main entries and 20-pound rails in the rooms. In the more recent operations 45- to 60-pound rails have been used on the main roads and 25- to 35-pound rails in the rooms. Most of the newer mines have adopted a track gauge of 42 inches.

Statistics for 1920 showed 345 shaft mines, 12 slope mines and 10 drift mines. The average depth of shaft was 225 ft. while the average slope length was 772 ft. The production from the different kinds of mines was: shaft mines, 69 004 807 tons; slope mines, 2 339 167 tons; drift mines, 717 096 tons. During the same year strip mines produced 367 009 tons or a little more than one-half of one per cent of the total production in the state.

Data from Illinois Coal Reports for the period 1899 to 1921

TABLE 1

KINDS OF HAULAGE IN ILLINOIS SHIPPING MINES

Used on Main Haulage System

Year	Locomotives*			Rope		Mules			Hand Tramming	
	No. Mines	Tonnage	No. Locomotives	No. Mines	Tonnage	No. Mines	Tonnage	No. Mules	No. Mines	Tonnage
1899	3	591 274		19	1 849 474	407	20 301 405	. . .	456	571 927
1900	7	1 331 579		27	2 424 278	512	20 410 169		374	987 570
1901	12	2 759 872		21	2 740 616	545	20 391 023		341	743 808
1902	13	3 043 497		15	2 626 182	289	22 050 515		7	30 329
1903	16	3 947 601		26	3 518 122	298	25 425 799		10	60 825
1904	22	4 779 822		28	4 287 836	302	24 829 705		18	107 426
1905	33	6 984 054		18	2 964 132	318	24 327 719		14	84 186
1906	46	8 726 686		30	2 789 640	324	24 371 623		12	75 547
1907	75	16 542 575	129	25	2 864 241	303	26 689 533		5	46 865
1908	88	19 024 665	185	32	3 273 753	283	25 182 634		4	28 678
1909	96	21 892 462	210	16	1 396 154	268	24 645 398	5600	4	24 548
1910	106	23 204 480	229	25	2 508 343	251	21 243 637		8	268 741
1911	137	29 310 173	303	24	2 321 016	219	16 839 883		7	287 585
1912	165	37 958 050	381	4	400 970	210	17 707 993	5744	1	29 682
1913	185	46 194 737	467	13	1 173 528	168	13 014 153	5372	5	132 298
1914	191	47 485 729	540	6	478 041	139	11 233 532	5244	1	106 858
1915	173	47 239 554	602	7	320 062	92	7 947 497	4265	6	502 645
1916	176	53 140 005	648	7	675 235	98	8 123 624	4213	3	343 272
1917	200	67 196 768	744	5	321 628	113	9 422 285	4913	6	471 373
1918	240	77 662 619	960	1	98 209	129	10 545 400	5065	0	. .
1919	259	66 686 930	1114	3	143 118	111	6 921 673	5370	0	
1920	276	66 441 191	1228	2	80 202	95	5 888 217	4974	0	
1921	280	71 329 567	1424	1	38 481	108	6 971 034	5348	0	

*Haulage by gasoline locomotives included as follows: 1907, 2–400 643 tons; 1908, 1–245 513 tons; 1909, 2–81 670 tons; 1910, 3–85 892 tons; 1911, 3–185 836 tons; 1912, 13–1 857 415 tons; 1913, 24–3 580 539 tons; 1914, 28–4 279 106 tons; 1921, 18–942 776 tons.

inclusive indicate a gradual reversal in prominence held by animal and locomotive haulage. (Table 1.) In 1899, 87.1 per cent of the tonnage in Illinois coal mines was handled by animal haulage. Locomotives hauled 2.5 per cent, ropes 7.9 per cent and tramming 2.5 per cent, but in 1921 it appears that both ropes and tramming were practically obsolete and that 91.2 per cent of the coal was moved by locomotives, and only 8.8 per cent by animals.

In the early '90's, several attempts were made to use electric locomotives, some of them meeting with considerable success; but 1899 was the first year in which an appreciable amount of coal was hauled by electric locomotives. Statistics for the number of electric locomotives in use prior to 1907 are not available, separate from the statistics for gasoline and other types.

Pertinent data on mine haulage were collected, in 1914, by the Illinois Coal Mining Investigation and published in Bulletin 13. Twenty-four typical mines that used mule haulage had average conditions as follows: daily coal production, 597 tons; weight of empty car, 1239 pounds; weight of coal per car, 2627 pounds. Similarly in 65 typical mines having mechanical haulage, the average statistics were: daily production, 1667 tons; weight of empty car, 1753 pounds; weight of coal per car, 4450 pounds. There were five mines using the rack-rail type of locomotive and seven using gasoline locomotives. Rope haulage was used in but six mines. All other mines were using trolley locomotives.

Table 2 gives a classification of the three chief systems of underground haulage in use throughout the state in the year 1921. Rope

TABLE 2

LOCOMOTIVE AND MULE HAULAGE IN 1921

System	Mines		Production		Ave. Tons Per Mine
	No.	Per Cent	Tons	Per Cent	
Mules only	109	33.2	9 976 493	12.9	91 527.5
Locomotives only	31	9.4	13 731 010	17.9	442 935.8
Locomotives for main haulage, mules and locomotives for gathering	189	57.4	53 598 971	69.3	283 592.4
Totals	329	100.0	77 306 474	100.0	234 974.1

haulage is not included because its use is very limited and the Coal Reports do not now segregate it. For the mines covered, this table shows how the haulage systems are related to production.

Statistics for the year 1921 covering 324 producing mines in 38 counties of Illinois show that electric haulage was used exclusively in but 31 mines or 9.6 per cent; mules performed all the haulage in 108 mines or exactly one-third; in the remaining 185 mines haulage was "mixed," that is, by both locomotives and mules.

III. THE SHAFT BOTTOM

8. *General Importance.*—The term "shaft bottom" applies to the portion of the mine that is contiguous to the bottom of the main hoisting shaft. It includes the terminal tracks for storing the loaded cars while waiting to be hoisted, the storage tracks for empty cars while waiting to be taken back to the working faces, and the necessary motor and supply rooms, foreman's office, pump rooms, run-arounds, shops and waiting rooms.

When it is considered that the shaft location may affect the haulage grades for the entire mine throughout the life of the mine, the importance of preliminary drilling to determine the contour of the coal bed is obvious, in order that the shaft bottom may be located as nearly as possible at the lowest point in the mine and the loaded trips hauled down-grade as much as possible. A shaft bottom on the loaded-car side should be either approximately level or at a grade of 1 to 1.5 per cent toward the shaft. The grades on the empty side of the shaft vary with the method of handling the empty cars.

An adequate shaft pillar should be provided about the shaft bottom to protect the shaft and the surface equipment from subsidence. In too many cases, however, where the original plans called for adequate shaft pillars, rooms have been started in the pillar in order to get coal quickly. In many cases it has proved very poor economy to mine out the coal too close to the shaft, for it should be remembered that this coal is not lost but merely deferred in its extraction to the time when the mine will be abandoned. Typical shaft bottom arrangements are shown in Figs. 12 to 18, inclusive.

The shaft bottom is the heart of the underground workings and is the busiest place in the mine. Here the loaded cars must be promptly hoisted or dumped and the empties returned to the working face to avoid blocking the traffic. In some mines from 1200 to 1500 cars are handled on the shaft bottom daily during an eight-hour shift, or an average of two to three per minute. The efficient operation of the whole mine, therefore, depends not only on shaft-bottom arrangement and mechanical equipment, but also on a proper balancing of the haulage from the various divisions of the mine to the shaft bottom which is the main terminal.

The first extensive use of self-dumping cages was in Illinois. At present they represent the prevailing method of hoisting coal, except in the longwall field. At a number of the older and smaller mines and very generally in the longwall field the platform cage is still used, the car being run off the cage at the surface to be dumped. In a few cases, two cars are placed on the cage platform for hoisting, either tandem or side by side. The speed of hoisting at the larger mines gives two to four hoists per minute. Mine cars vary in capacity from two to four tons each.

The chief items to be considered in the shaft-bottom layout are:

Arrangement of tracks to permit the locomotive to land a loaded trip and to obtain an empty trip without delay, so as to prevent interference of one locomotive with another.

Storage space for loads and empties.

Shaft-bottom grades.

System of handling loads and empties, including caging, if the cars are to be hoisted.

Arrangements for safely receiving the men who have been lowered; also adequate waiting room for men who have gathered on the shaft bottom previous to being hoisted to the surface.

Suitable arrangements for handling equipment and supplies, such as timber, oil, waste rock, sump coal, and broken cars.

Conveniently located mine manager's office, locomotive barns, repair and supply shops, pump-rooms and mule stables.

The act of haulage is really completed when the car is placed on the cage ready to be hoisted, but often haulage and hoisting data are not kept separately. Only data upon hoisting, ventilation, and such collateral topics as have an effect upon haulage performance are considered in this discussion.

9. *Delivering Cars to Shaft Bottom.*—It is important that the main-line locomotives be able to land the loaded trips at the bottom and take up the empty trips for the return with the least possible interruption. The likelihood of interference increases with an increase in the number of locomotives hauling to the shaft bottom. With two locomotives, one coming from each side of the shaft, there should be no interference and no delay, provided there is ample storage for

empty cars. Where two or more locomotives come to the shaft bottom over the same route, interference on the shaft bottom between the incoming and outgoing locomotives is probable unless a definite schedule is maintained and proper provision is made in the shaft-bottom layout. Three different ways of preventing such interference, described later in detail in connection with the several shaft-bottom arrangements herein given, are as follows:

(1) Adequate length of double track in each direction from the shaft on the main haulage road, as described under Mine A, p. 56.

(2) Separate outlets from the shaft-bottom empty-storage track to the several sections of the mine, with grade track crossings eliminated by the use of cross-over bridges, as described under Mine C, p. 58.

(3) A trip despatcher or haulage boss on the shaft bottom who is in touch by telephone with flagmen at the junction points, and thus directs the incoming trips.

10. *Storage Space for Loads and Empties.*—Adequate storage tracks for loaded and empty cars, and a suitable arrangement of such tracks and their approaches should be provided, as these items very largely determine the regularity and continuity of cars supplied to the cager for hoisting. A shortage of railroad cars on the surface or an accident in the shaft may cause delay in hoisting; therefore, the shaft bottom should provide adequate storage and flexibility in handling cars and incoming trips.

Data in Table 3 show variations in storage capacity at a number of mines studied, and Figs. 12 to 18 show a number of different arrangements of storage tracks. In Table 3, ''Storage capacity loads'' means the number of cars that can be stored on the track from the shaft to a point where the incoming locomotive ordinarily is cut off from the loaded trip; and ''Storage capacity empties'' means the number of cars that can be stored on the empty-car track without interfering with the caging operations or with the passage of the incoming locomotive. Any extension of storage space that interferes with regular operations should not be included as regular storage capacity.

Although the storage capacity on the shaft bottom is figured for a certain number of cars, the varying sizes of trips and times of

TABLE 3

SHAFT BOTTOM DATA, INCLUDING LABOR COSTS

Costs Figured for Daily Averages, Regular Working Force

Mine	A	B	C	D	E	F	G	H	I	J	Ave.
Ave. Daily Tonnage	4500	5200	4500	5000	3800	3600	2600	3400	4000	3200	3980
Ave. No. Hoists, Hour	140	155	110	180	200	105	112	130	167	134	143.3
Total No. Cars Hoisted	1125	1245	860	1440	1600	800	800	1050	1330	1070	1114
Total No. Cars in Use	490	625	510	640	580	450	545	485	600	500	542.5
Ave. Wt. Coal per Car	4	4.3	5.25	3.5	2.3	4.5	3.3	3.25	3.0	3.0	3.64
Storage Capac. Loads	100	100	100	60	50	60	92	100	50	120	83.2
Storage Capac. Empties	160	52	160	60	50	60	72	100	50	90	85.4
Method of Handling Cars:											
Loads	Aux. loco.	Grav.	Grav.	Grav.	Grav.	Aux. loco.	Grav.	Grav.	Grav.	Grav.	
Caging	Auto	Auto	Auto	Auto	Auto	Auto	Auto	Auto	Auto	Auto	
Empties	Aux. loco.	Grav.	Grav.	Car lift	Car lift	Aux. loco.	Grav.	Grav.	Grav.	Grav.	
Employees:											
Cagers	1	4	1	3	3	1	2	1	1	1	1.8
Spraggers and Blockers	3	6	2	2	3	3	3	2	3	4	3.1
Couplers	2	2	1	2	2	2	1	2	1	1	1.6
Car Distributors and Switchers	1	1	1	1	3	1	1	1	1	1	1.2
Total No. Employees	7	13	5	8	11	7	7	6	6	7	7.7
Total Daily Wages	51.00	95.25	36.50	58.75	82.50	51.00	51.25	43.75	43.75	51.00	56.28
Shaft Bottom Labor Cost in Cents per ton	1.13	1.83	0.81	1.18	2.17	1.42	1.97	1.29	1.09	1.59	1.42

NOTE: General items, as Oiling, Cleaning, Sump, etc., are not charged to costs.

arrival often prevent the rated capacity from being available. The location of the connections between the main bottom tracks and the run-around tracks, often called "the motor runs," and the points where the locomotives are cut loose from the trip determine the storage capacity of the shaft bottom to a great extent. For example, this cut-off point may be so located that when one locomotive follows another into the bottom on the same side, the second locomotive will be delayed until the last loaded car of the first trip has passed the entrance to the motor run, and the first locomotive will be delayed until the second loaded trip has cleared the junction point between the loaded and empty tracks on the main entry, unless there is a double track on the main haulage road.

11. *Handling Cars on Shaft Bottom.*—There are three distinct operations in connection with the handling of cars on the shaft bottom:

(1) Delivering loaded cars to the cager after the main-haulage locomotive has been cut off.

(2) Caging.

(3) Taking empty cars from the cage to the empty storage track.

Delivery of Cars to Cager

There are three methods by which the loaded cars after being cut off from the locomotive may be delivered to the cage; first, by pushing and spragging, second, by car haul, and third by a small locomotive running on a center track.

(a) When the control of the mine car after the locomotive has been cut off is left entirely to the spragger, the grade toward the shaft is usually about 1.5 to 2 per cent from the locomotive cut-off point to a point about two car lengths from the cage, and from this point on to the shaft the grade is increased to about 3 per cent so that the loaded car may have sufficient impetus to bump the empty car off the cage. Too steep a grade on the shaft bottom is dangerous for the spraggers and switch-throwers.

If there is a slight up-grade on the approach to the shaft bottom so that the locomotive must continue pulling until a cut-off switch is reached, such a switch should be automatically thrown by the locomotive. If the speed at which the trip is cut off is excessive there is danger of the cars getting beyond control. If the same employees always handle the loaded trips, they become skilled in their work and can accurately judge the distance the cars will run and the number of sprags necessary, so that very few run-away trips occur although this method of handling cars is extensively used. Handling by gravity and spragging is a continuation of the method employed when the cars were much smaller than those commonly used now. The present tendency is to install heavier equipment both in mine cars and in locomotives so that the problem of controlling the cars by hand under such conditions is much more difficult than formerly.

Several types of sprags are shown in Fig. 5. The ordinary double-cone spoke sprag *a* is thrust between the spokes of the moving wheels, thus causing the wheels to slide on the rail; the block sprag *b* may be placed on the rail in front of the wheel, or it may have a flat face cut out to fit the flange of the wheel *c*, *d*, and be placed in front of the wheel. The block form gives much greater surface of contact than the cone type and one block is as effective as several cone sprags. On account of the smaller number of block sprags required, there is also a saving in time in the application of sprags.

(b) A car haul consists of an endless chain to which are attached at regular intervals ''catches'' that engage the axles of the cars and push the latter forward toward the cage. A similar device may be used for moving the empty cars on the opposite side of the cage.

(c) By means of a relatively small locomotive running on a third or center track (Fig. 6), and provided with an arm that can be moved in and out transversely on either side of the locomotive, cars on either track are pushed forward toward the cage. The advantages of this system are that at all times the cars are under control, and they may be moved in either direction as desired, the safety on the bottom being thus increased. A car-haul can control the movement of cars for the length of its construction only—perhaps 75 feet—whereas an auxiliary locomotive will regulate the travel of the loads or the empties for the entire length of the bottom with the exception of about 50 feet on either side of the shaft. Another fact in favor of the auxiliary locomotive is that it proves useful in replacing derailed cars anywhere on the shaft bottom.

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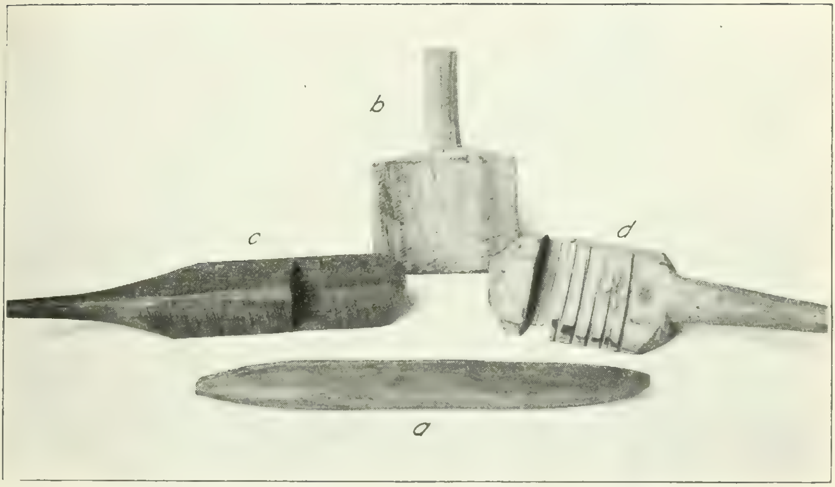

FIG. 5. TYPES OF SPRAGS

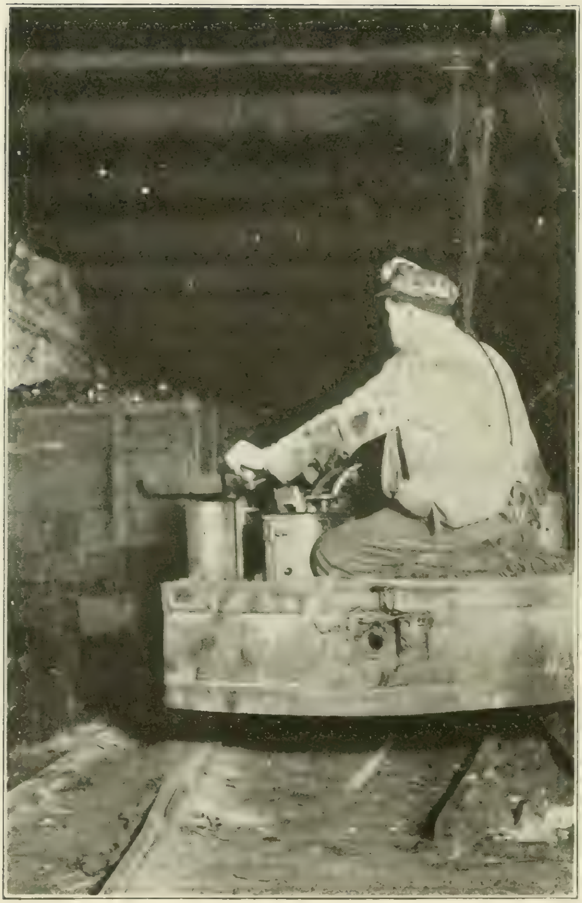

FIG. 6. CENTER-TRACK PUSHER LOCOMOTIVE

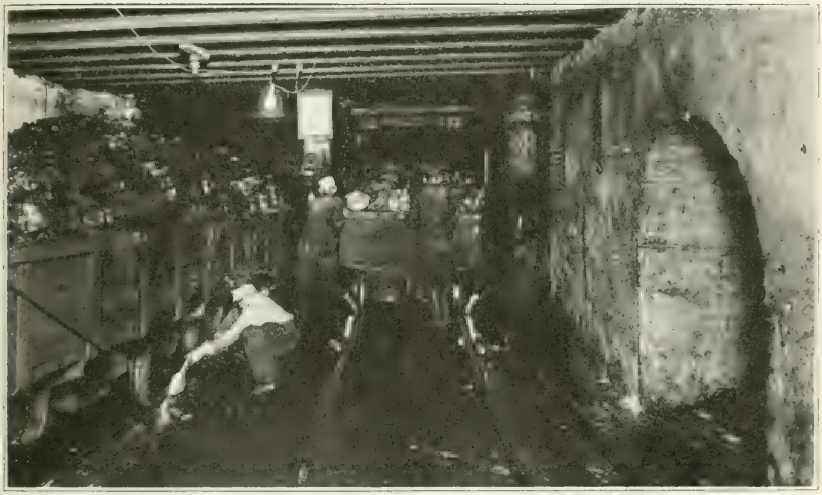

FIG. 7. AUTOMATIC CAGING DEVICE AND USE OF SPRAG

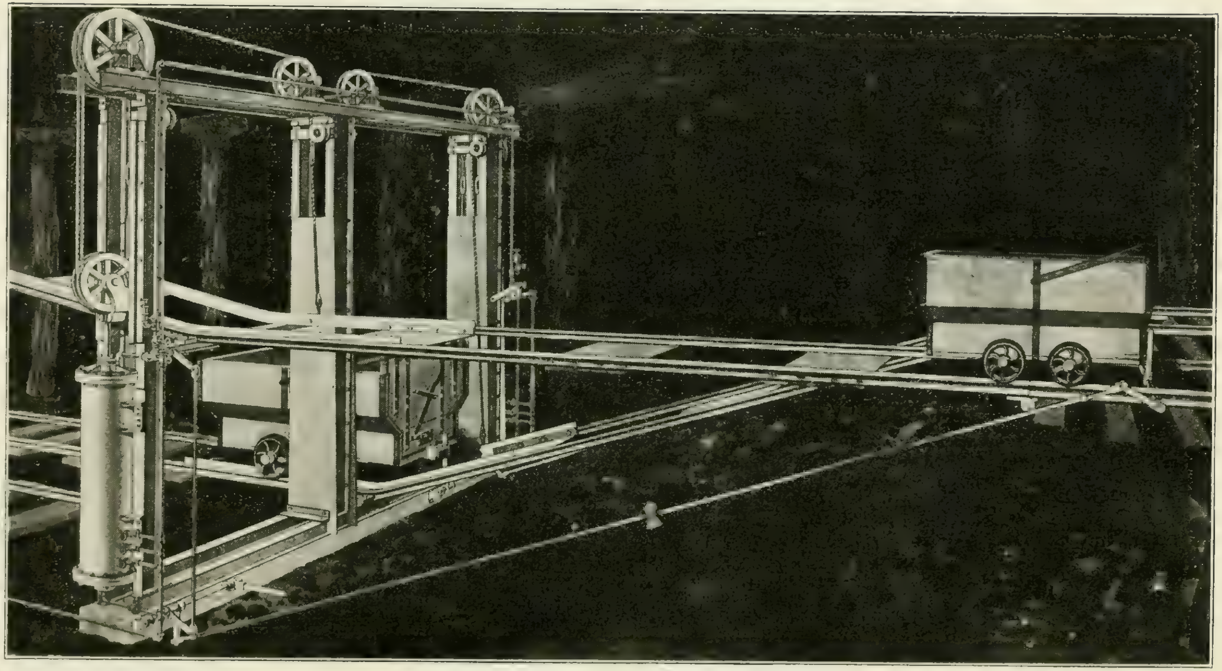

FIG. 8. CAR LIFT

Caging

Cars are caged by hand or by an automatic caging device, the loaded car bumping the empty car off the cage. Fig. 7 shows an automatic caging device used at most of the newer mines. A pair of dogs nearest the shaft is opened automatically when the cage strikes the bottom, thus permitting the loaded car to be pushed upon the cage. At the same time a second pair of dogs, farthest away from the shaft, is thrown across the track and stops the incoming car. As the cage rises off the bottom, the dogs that were across the track open and the other pair fall back over the track, thus permitting the loaded car to be pushed forward ready to be put on the cage when it next descends. In some instances, caging is carried on so rapidly and with such precision that the signal to hoist is given before the car has come to rest on the cage.

Removal of Empty Cars

Owing to unfavorable natural conditions it is often necessary to do considerable grading in order that an empty car may run by gravity from the cage to the empty-storage track. An arrangement often used when the cars are caged from one side only is to have the track leading from the cage terminate in a "kick-back" which gives the empty cars sufficient impetus to cause them to run by gravity to the empty-storage tracks, where they are formed into trips. By means of a mechanical car lift (Fig. 8) the empty car may be raised 8 to 12 feet and thus, in running down a grade, be given an impetus that will cause it to run by gravity directly to the empty-storage tracks; or from the car lift it may go first to a "kick-back" and thence to the storage track.

A three-track arrangement with an auxiliary locomotive operating on the center track, similar to that described as being used on the loaded side of the shaft, has many advantages for handling heavy equipment and gives a very flexible method of operation. A greater length of shaft bottom on the empty side is necessary for this arrangement but it provides increased storage space for empty cars and also a convenient way for shifting broken cars.

12. *Handling Men on Shaft Bottom.*—The shaft-bottom arrangements for handling men depend upon whether the hoisting shaft for coal is used also for hoisting men, or whether an auxiliary shaft is

used for men and materials. According to the present agreement in Illinois between the mine operators and the United Mine Workers of America, the men are hauled to and from the shaft bottom and inside partings of the mine. Consequently, greater numbers of men may be expected to congregate at one time on the bottom than was the case when the men walked to and from their work. This condition should be taken into account in the arrangement of the shaft bottom.

When the men are hoisted at the main hoisting shaft it is common practice to run a man cage about nine o'clock in the morning, one or more during the noon hour, and one during the afternoon when the shot firers enter the mine. These are in addition to the cages at the regular morning and afternoon lowering and hoisting times. The activity on the shaft bottom during the working hours makes traveling dangerous, and in a number of mines special traveling ways are provided to the waiting rooms required by the Illinois Mine Law so that men are kept away from moving cars.

The approach to the hoisting shaft and to the escape-way at the air shaft should be carefully chosen and easy of access. The waiting rooms are usually so located that, in passing to the cage, the men pass the ''checking'' room and turn in the checks given them on entrance in the morning. At one mine a waiting room has been made by placing flooring about seven feet above the main tracks and providing seats in the room thus made. Such an arrangement is possible of course only when there is unusual headroom on the bottom. Such gathering places for men offer an opportunity for the display of safety signs and pictures. Indeed, moving pictures relating to safety might also be shown while the men are waiting to be hoisted, though no instance of this being done is on record.

13. *Handling Supplies, Equipment, and Refuse.*—The problem of handling equipment, supplies, broken cars, etc. is most successfully solved where there is a separate man and materials shaft, which is usually the air shaft also. The mines provided with separate hoists at the air shaft have this advantage also that all refuse can be hoisted and taken to the dump pile without either interfering with the hoisting of coal or requiring any changes of chutes in the tipple, as is necessary when the same self-dumping cages are used both for coal and rock.

14. *Handling of Sump Coal.* The method generally employed for removing the coal that falls into a sump is to have it hand shoveled into a mine car. In addition to the inconvenience of this method there is a certain amount of danger attached to it, due to unintentional lowering of the cage upon the man in the sump or from objects falling down the shaft. One solution of this problem is a track laid into the sump under the cages at right angles to the cage tracks. Two mine cars are run into this sump, one under each cage. When they become full of coal they are withdrawn and replaced by empty cars. Such an arrangement is possible only where a crosscut or entry on the cage landing opens at the end of the shaft, and where the conditions are such that suitable grading may be done in order that the cars may be hauled from under the cage.

At the Bunsenville Mine of the U. S. Fuel Company, provision has been made whereby cars may be run under the cage landing and there loaded from a hopper with a drop-bottom attachment. These cars are then pushed to an electrically operated auxiliary hoist and hoisted a distance of 13 feet to the shaft-bottom level.

At some mines a removable box with a drop bottom or side has been placed in the sump and fitted into the guides so that when full of coal it can be attached below the cage and hoisted the height necessary to permit the contents to be discharged through a detachable chute into an empty car on the shaft bottom.

15. *Arrangement of Offices, Stables, Shops, and Supply Rooms.*—At many mines greater attention could advantageously be given to the provision of larger and more adequately equipped mine manager's offices on the shaft bottom, where managers and their assistants may meet for consultation.

Where mules are used they are generally stabled underground near the shaft bottom. The construction of underground stables has been specially provided for in the Illinois mine law, which specifies a separate air split, fire-proof construction throughout, automatic sprinklers, fire-proof doors, covered bins, and covered cars for hay and grain. The worst accident in the history of Illinois mining, the Cherry mine fire, was due to the careless handling of hay. The standard stable of one large company operating in Saline County is shown in Fig. 9. The construction of this stable is fire-proof throughout, consisting of steel roof support, full-height concrete walls and concrete floors, stall partitions, feed boxes, feed bins, and harness rooms. The stable feed bins and harness rooms are fitted with steel doors. Separate hay and grain boxes are provided for each stall, with one water trough for two stalls. An automatic sprinkler system is installed directly over the feed boxes. The stall partitions are built of concrete 42 inches high, topped with a wire screen 24 inches high. Hooks are provided at each stall for holding the harness when not in use. A track in the center of the stable is used for handling supplies and loading out manure. Additional space is provided for washing the mules. Drainage is provided by a tile conduit extending under the full length of the stable. Every Saturday the stable is thoroughly washed out with a hose and thus maintained in a sanitary condition.

The central point for storing locomotives over night or during idle periods should be readily accessible from the different sections of the mine. The locomotives are left standing along the main tracks with the trolley poles down, if no barns are provided for their storage. Where storage-battery locomotives are used, provision is made for charging stations and these are usually installed in a special locomotive barn.

In connection with locomotive haulage, it is becoming more and more common to provide on the shaft bottom a fairly complete repair shop in which there are often one or more motor pits. Moreover, time might be saved where gathering locomotives are used, by establishing at central points in the inner workings auxiliary repair shops fitted with motor pits for minor repairs. This has been done at one mine in Saline county in connection with an underground sub-station. For line repairs and bonding of the rails and also for certain minor repairs to locomotives, a specially-equipped portable repair car may be maintained.

Usually broken cars are hoisted and taken off at the ground landing to be repaired in the repair shop on the surface; but at a few mines provision has been made for making small repairs to mine cars underground, particularly to the running gears, couplings, drawbars, etc. A repair room for this purpose is sometimes located near to and connected with the empty-storage track.

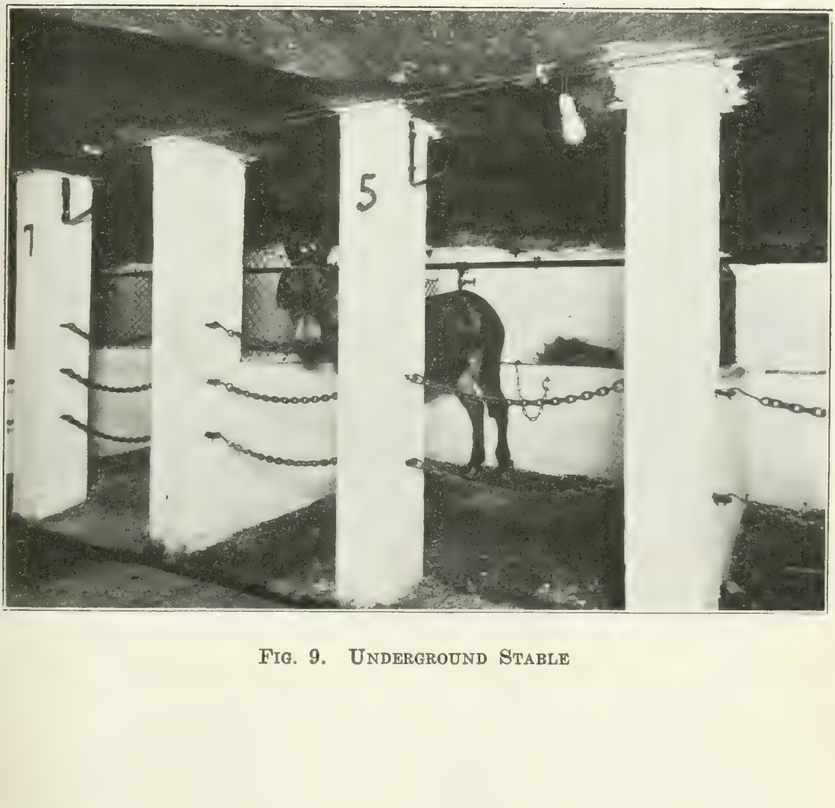

FIG. 9. UNDERGROUND STABLE

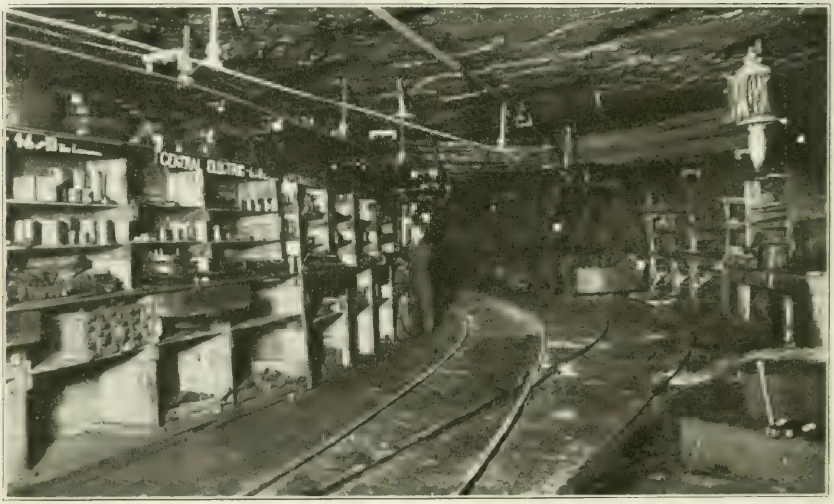

Fig. 10. Underground Supply Room

The following is a report of locomotive repair items for one day at one of the larger mines in Illinois where 7 main-line and 19 gathering locomotives are used.

Main-Line Locomotives	Nature of Repairs	Time Spent in Repair Shop	
No. 21	Arc lights	0 Hr.	15 Min.
21	Suspension bar down	0	55
		—	—
Gathering Locomotives		1	10
No. 26	Reel ball race	2	30
9	Reel circuit ground	0	15
8	New trolley pole	0	10
16	Reel stud broken	0	30
23	Reel resistance	0	52
26	Short circuit	0	21
24	Lead off resistance	0	12
7	Sand-rod broken	0	8
22	New reel armature	0	20
4	New reel armature	3	0
5	Lead blown off reel motor . . .	0	32
12	New trolley pole	0	15
5	Resistance blown up	1	0
		—	—
		9 Hr.	47 Min.

Some companies maintain near the shaft a room well equipped with supplies needed in connection with the operation and repair of mining machines and locomotives. (Fig. 10.) Such a supply room, usually in charge of a storekeeper who checks out materials by a system similar to that generally used on the surface, aids materially in keeping account of the repairs upon each mining machine or locomotive. Oil and grease are sent underground in barrels and are usually stored in an offset to the empty run-around near the oiling station. On account of the fire risk special precautions should be taken when handling and storing this material. Considerable sand is used daily in some of the mines, at one mine eight tons per day being used for sanding the rails. The usual method of handling the sand is to dry it on the surface and then send it below in mine cars for distribution to central points. Sometimes a pipe through a bore-hole from the surface carries the sand to a central distribution point near the shaft bottom.

16. *Shaft-Bottom Support.*—In some mines where there are favorable natural roof conditions and an ample height of coal, very

little support to the top and sides is necessary; but in most mines a large amount of roof and side support must be used. Much greater permanency now marks shaft-bottom construction than formerly and in many of the more recent shaft bottoms concrete arches or steel I-beams, with wood lagging or concrete roofing between the beams, have been installed on the shaft bottom as part of the initial development. Concrete sides serve the double purpose of sustaining the roof supports and the coal ribs.

Three types of permanent construction are shown in Fig. 11. In the first type (*a* and *b*) structural steel is used for the posts and the caps, with plank lagging on walls and roof. In the second type (*c* and *d*) concrete is used for the walls, structural steel for the caps, and the lagging is either plank or corrugated or sheet iron. In the third type (*e*) concrete is used exclusively for the walls and the roof, the roof being an arch.

Cost figures for these three general types of support have been furnished by Allen & Garcia, Chicago, Illinois, the estimates being based upon the average cost of the various materials in place as of August 1, 1921, and upon sets being placed at 4-feet centers. The constructions illustrated are calculated to withstand top pressures of 750 pounds per square foot and side pressures of 500 pounds per square foot. Concrete is estimated as costing $30 per cubic yard; structural steel, 8 cents per pound; iron, 10 cents per square foot; and lumber, $65 per thousand board feet.

For a shaft bottom or double-track entry, Fig. 11*a*, using 6-inch H-beams for posts and 12-inch I-beams for caps, the cost was approximately $24 per lineal foot.

For a single-track entry, supported exclusively by structural steel, Fig. 11*b*, 6-inch H-beams are used for both posts and caps. The estimated cost per lineal foot was approximately $18.

In the second general type of construction, for both single-track, Fig. 11*c*, and double-track, Fig. 11*d*, with walls 18 inches thick at the bottom and 12 inches thick at the top, and using for the narrow entry 6-inch I-beams for caps, and for the double-track entry 12-inch I-beams, the cost per lineal foot in the two widths of entry was respectively $25 and $30.

For the third type, Fig. 11*e*, in which concrete is used exclusively with walls of the same thickness as in the preceding type and the

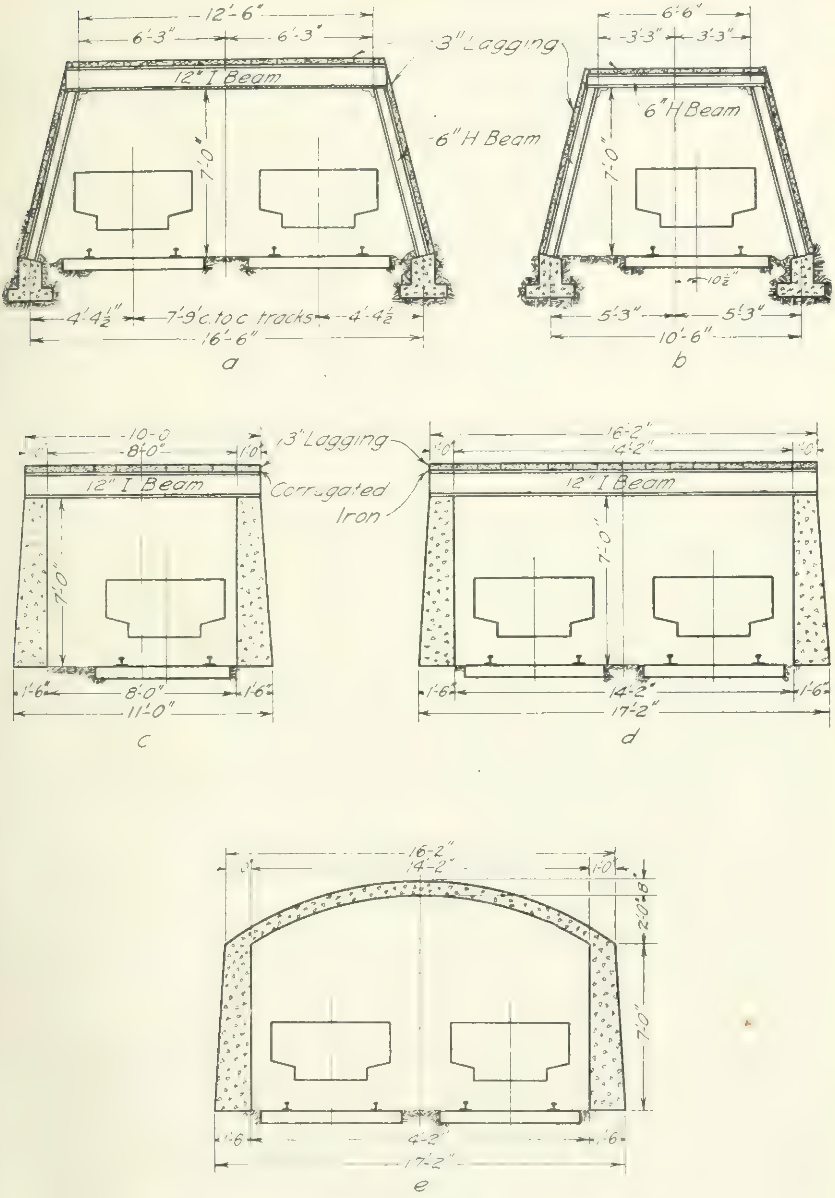

FIG. 11. TYPES OF PERMANENT SHAFT BOTTOM SUPPORTS

arch uniformly 8 inches thick, the cost per lineal foot was approximately $31 for the dimensions given.

A coating of cement put on roof and ribs with a cement gun is being extensively experimented with in an effort to prevent spalling off of the coal. This cement coating should not be applied until the roof and ribs have been thoroughly brushed or cleaned to remove all dust and loose fragments of coal, thus ensuring a solid foundation for the cement, which would otherwise spall.

17. *Typical Shaft-Bottom Plans.*—Typical shaft-bottom plans for several Illinois mines are shown in Figs. 12 to 18, inclusive. A characteristic feature of the bottoms in most of the newer mines in Illinois is that the shafts are in the shaft pillars off from the lines of main haulage and the tracks leading to and from the shafts are approximately at a right angle to the main haulageways. This is illustrated in Fig. 13 and is commonly known as the ''A'' type of shaft bottom. If the empty tracks leading from the back of a shaft to the main haulageway are not parallel to the incoming loaded tracks, but at an angle of 30 deg. to 45 deg., as shown in Fig. 15, the bottom is said to be of the ''V'' or triangular type.

Data on the general layout, operation and cost of operation for ten mines are given in Table 3. A detailed description of the particular features of the plans and the methods of operation follows. The term ''bottom men'' as used in this bulletin includes the men engaged in handling the loaded cars, i.e., cagers, spraggers, switchers, and couplers but not the oilers and sump men who work on the bottom, but do not handle the cars. The costs are based on the 1920 wage scale as follows: motormen, $7.50 per day with an additional allowance for hauling men to and from the partings, making the wage about $8.03 per day; trip riders and cagers, $7.50; couplers, switchers, spraggers, $7.25.

Mine A

This shaft bottom (Fig. 12) has a three-track arrangement on each side of the shaft. The main-line locomotive, upon reaching the shaft bottom with a trip of loaded cars, is stopped at the point *a* and the locomotive cut off. A ground switch is thrown by the trip rider who then gets back upon the locomotive, which proceeds to the empty storage track. A six-ton auxiliary locomotive (Fig. 6), which is on

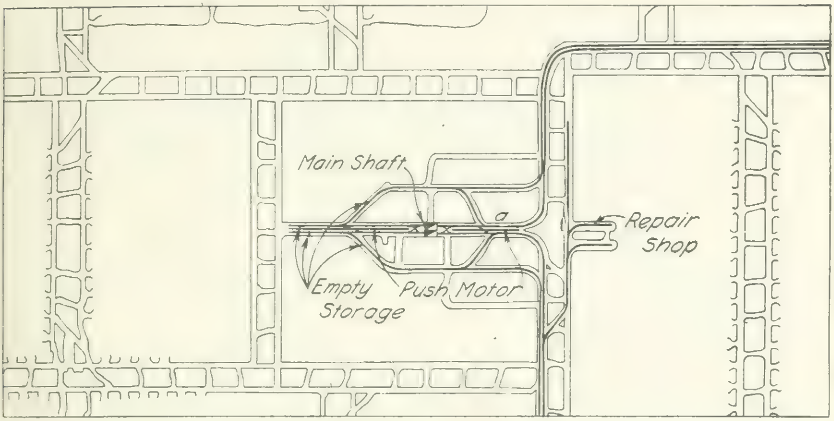

FIG. 12. MAP OF SHAFT BOTTOM—MINE A

the middle track and is provided with an extension arm, then moves the trip under complete control to the shaft where it is caged by an automatic cager.

On the empty-car side of the shaft another six-ton locomotive with a movable arm collects the empty cars as they come from the cage and places them on the empty-storage tracks. In this mine, 14 locomotives haul directly from the working face to the shaft bottom, seven coming from the north and seven from the south side of the mine. At the same mine, the double track extends for 2500 feet on the main haulage entry in each direction from the shaft bottom. This permits the locomotives to proceed on their return empty trip without interruption from the incoming loaded trips. This double-track arrangement also permits the entire number of locomotives if necessary to concentrate near the shaft bottom with loaded trips, giving in effect a very large loaded-storage capacity which may include every car in the mine without interfering with the empty return tracks. The empty-car storage shown in Fig. 12 is ample for ordinary operation of the mine and provides for the storage of about 45 cars on each side of the shaft. Delays on the bottom at this mine are small although an average of 1125 cars are hoisted daily, the empty cars weighing 2750 pounds and holding 4 tons of coal. The shaft-bottom force handling cars includes 1 cager, 3 spraggers, 2 couplers and 1 car dis-

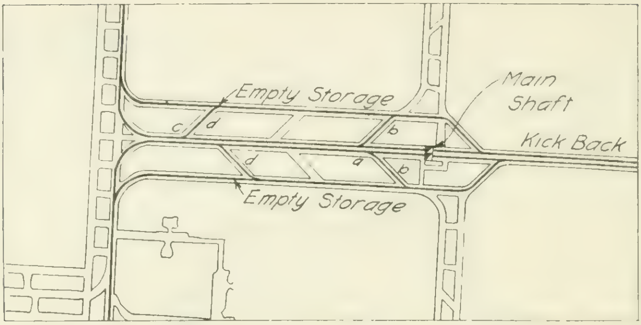

FIG. 13. MAP OF SHAFT BOTTOM—MINE B

tributor, at a total daily wage of $51 according to the 1921 wage scale, or 1.13 cents per ton.

Mine B

On this shaft bottom (Fig. 13) two tracks lead to and from the shaft. The main-line locomotives usually cut off from the trips at a point *a* about 100 feet from the shaft and, after passing through a switch that is automatically thrown, proceed through the motor run cross-cut *b*. From the point *a* to the automatic cagers at the shaft the loaded cars are controlled by spraggers. The empty cars are run by gravity to a kick-back and thence to the empty-car storage track where they are formed into trips ready for the locomotives that come through the motor run *b*. Of the six locomotives that come to the shaft bottom, four are of the 15-ton type and are used for main-line haulage only, while the remaining two, which are of the 8-ton reel-and-trolley type, are used for gathering as well as for main-line haulage.

From 1200 to 1500 cars are caged per day on this bottom. Occasionally there is some congestion when the trips reach the bottom in rapid succession, due to lack of empty-storage space and a single track on the main haulage roads. This congestion could be obviated by double-tracking the main haulage roads for a distance of 200 to 300 feet inbye from said junction with the empty-storage tracks, and by increasing the empty-storage trackage. This could be accomplished by cutting off the locomotive at the point *c* and having it go through the cross-cut *d*, the loaded cars being controlled from *c* to the cager by spraggers.

The average daily tonnage hoisted at this mine is 5200 tons or 1245 cars, each holding 4.3 tons. The average hoists per hour are 155 and the bottom employees are 4 cagers, 6 spraggers and blockers, 2 couplers, and 1 car distributor and switcher. The total daily wage according to the scale prevailing in 1921 was $95.25, giving a shaft-bottom labor cost per ton of 1.83 cents.

Mine C

The bottom arrangement (Fig. 14) provides for a separate haulage way to each of the four sections of the mine 1, 2, 3 and 4. There are two tracks on each side of the shaft, and after the locomotive is cut off at the point *a* the cars are moved to the shaft by spraggers and automatic cagers. The main-line locomotives approach the inbye end of the shaft bottom, *b*, by different routes, but all of them are detached from the loaded trips at the point *a* and pass through the motor runs, *c*, to the empty storage tracks. At two places, *e*, where crossings are necessary, overhead bridges permit the loaded trips

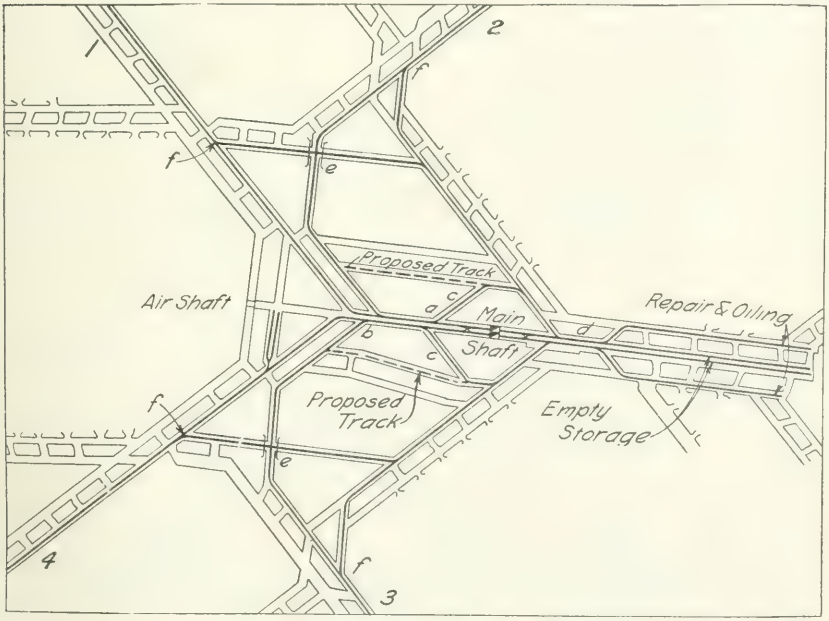

FIG. 14. MAP OF SHAFT BOTTOM—MINE C

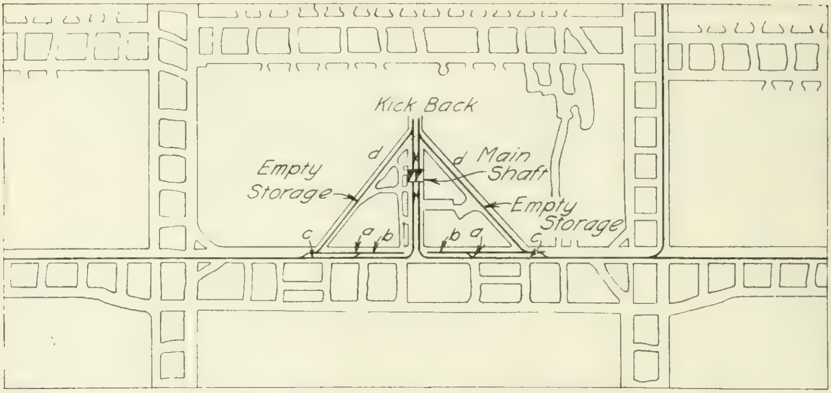

FIG. 15. MAP OF SHAFT BOTTOM—MINE D

to pass over the empty trips. From the junction points, *f*, of the loaded and empty tracks the haulage roads inbye are single track. The two "proposed tracks" parallel to the main bottom were intended as an extra locomotive run-around for sections 2 and 3, but they have not been needed to date.

The empties run by gravity from the cage to the empty storage track, *d*, where they are coupled to the empty trips.

With this arrangement a daily output of 4500 tons or 860 cars, each holding 5.25 tons of coal, is handled with 1 cager, 2 spraggers, 1 coupler, and 1 switcher, at a daily labor cost of $36.50 or 0.81 cents per ton.

Mine D

The roads leading to the shaft bottom (Fig. 15) are single-track and the locomotives are cut off along the main entry at *a* and, by flying a switch, run upon the parallel side track *b*. After the trip has passed the motorman brings the locomotive up behind the trip and pushes it to the automatic cager if this be necessary, or the trip may have sufficient momentum to run to the cager and may have to be controlled by sprags. The locomotive backs to the junction joint in the empty-storage track, *c*, and there picks up the empty trip. A mechanical car-lift and kick-back sends the empties by gravity to the empty-storage track, *d*.

Two locomotives operate in each of the east and west sections of the mine. Movements of trips to and from the shaft bottom are controlled by telephone communication from the several partings to the haulage boss who knows that the road is clear before giving the right-of-way; thus only one locomotive from each section is permitted on the shaft bottom at one time.

Five thousand tons or 1440 cars per day are handled on this bottom by 8 men, 3 cagers, 2 spraggers, 2 couplers, and 1 switcher, at a total labor cost per day of $58.75 or 1.18 cents per ton.

Mine E

This shaft bottom is that shown in Fig. 16. The locomotives are detached from the loaded trips on the main entries, the loaded cars proceeding by gravity to the shaft under control of sprags. Caging is done by hand. The empty cars are elevated by a mechanical car-lift and run by gravity from a kick-back switch to the empty-storage track. The daily output is 1600 cars or a total of 3800 tons. The shaft-bottom force includes 3 cagers, 3 spraggers, 2 couplers, and 3 switchers at a total daily labor cost of $82.50, or 2.17 cents per ton.

Mine F

The shaft-bottom arrangement is similar in general to Mine A, but the loads are pushed by ten-ton locomotives operating on the

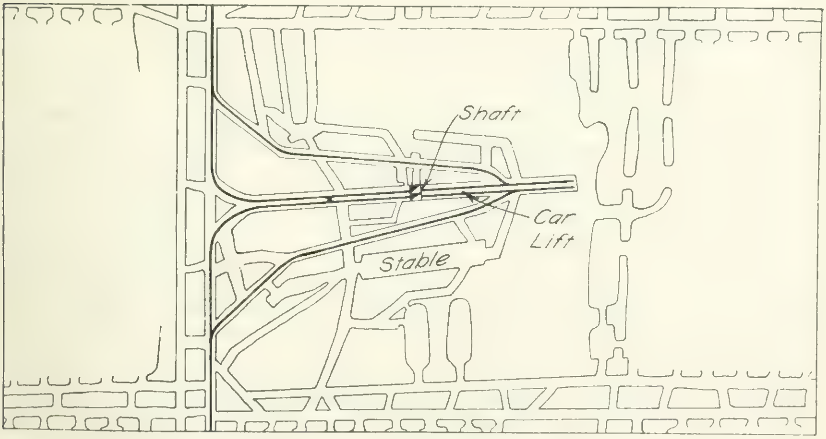

FIG. 16. MAP OF SHAFT BOTTOM—MINE E

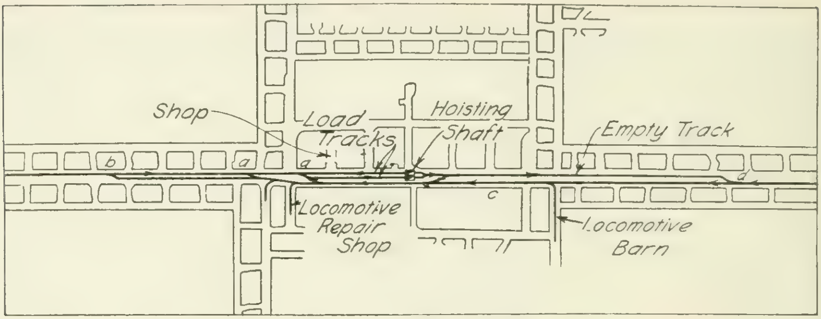

FIG. 17. MAP OF SHAFT BOTTOM—MINE I

middle of three tracks instead of by a six-ton locomotive as in Mine A. Only two locomotives come to the bottom, one from each side of the mine, and a combined main and gathering system is used; whereas in Mine A the locomotive hauls directly from the working face to the bottom, the trains consisting of from 10 to 20 cars. The empty cars are elevated by an electrically operated drag-line and then run by gravity to the empty-storage track.

With the present arrangement, and hoisting per day 800 cars that hold 4.5 tons each, the labor force is 1 cager, 3 spraggers, 2 couplers, and 1 switcher. The total labor cost is $51.00 per day, or 1.42 cents per ton.

Mine G

The shaft bottom is triangular in shape, similar to that shown in Fig. 15. There are two tracks on the shaft bottom and cars are controlled by spraggers after the locomotive is cut off. On the main east approach there is a slight up-grade, and a small electric drag-line is employed to pull the loads a short distance upon the main track. An automatic cager is used and the empty cars run from the cage to the empty-storage track by gravity. The location of the motor run is similar to that in Mine B and the empty-storage tracks extend beyond the shaft, as in Mine C.

For an output of 2600 tons per day, or 800 cars of 3.3 tons capacity, the labor force is 2 cagers, 3 spraggers, 1 coupler, and 1 switcher, at a labor cost of $51.25 per day or 1.97 cents per ton.

Mine H

This shaft bottom is similar to Fig. 13. The daily output of 3400 tons is handled in cars holding 3.25 tons each, by 1 cager, 2 spraggers, 2 couplers and 1 switcher, at a total shaft bottom labor cost of $43.75, or 1.29 cents per ton.

Mine I

This shaft bottom, Fig. 17, differs from the A or V type commonly used in Illinois, as the hoisting shaft is in line with the main entries that extend east and west from it. Coal is hauled to the shaft bottom from both directions, but caging is done from the west side of the shaft only. The locomotives are detached from the loaded trips from the west at one of two points *a* and obtain their empty trips at *b*. The loaded trips from the east are pulled past the shaft on track *c* and backed in on the loaded tracks at *a*. The locomotives that are hauling to the east side go along *c* to the entrance to the empty storage, *d*, to obtain their empty trips.

The repair shops are conveniently located on the west side of

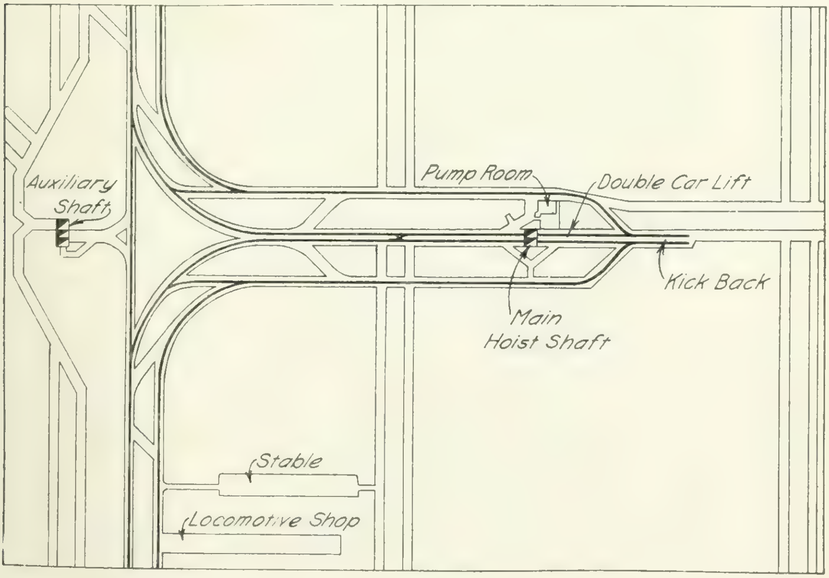

FIG. 18. MAP OF SHAFT BOTTOM—MINE J

the shaft, and the motor barn in which is the charging station for the storage-battery locomotives, on the east side of the shaft. A switch is provided near the shaft for sidetracking cars for oiling.

For an output of 4000 tons per day, or 1330 cars each holding 3 tons, the labor force is 1 cager, 3 spraggers, 1 switcher, and 1 coupler, and the total cost per day $43.75 or 1.09 cents per ton.

Mine J

Fig. 18 is a sketch of a shaft bottom somewhat similar to that in Mine E. Here, however, the entries are parallel. The empty cars can be hauled out along either of the main entries to the north or to the south. The daily production averages 3200 tons and is hoisted at the rate of 134 cars per hour with a shaft-bottom force of 1 cager, 4 spraggers, 1 coupler and 1 switcher, at a daily labor cost of $51.00 or 1.59 cents per ton.

18. *Shaft-Bottom Delays.*—At one mine a detailed study of delays on the bottom was made for one day, and the results are plotted in Fig. 19. Starting at 7:00 A.M., as shown by the diagram, there were 78 loaded cars on the bottom ready to be hoisted. There were also 8 empties. The loads were hoisted by 7:40 but the first trip did not reach the bottom until 7:57, thus causing a delay of 17 minutes. The diagram also shows delays in hoisting extending from 8:20 to 8:30; 8:50 to 9:00; 11:11 to 11:15, due to no cars being received on the bottom. Eight times during the forenoon—at 8:05, 8:09, 8:35, 9:07, 9:15, 9:31, 9:40 and 9:56—the diagram shows that the incoming trips reached the bottom just as the last car was hoisted, thus probably causing a slight slowing up in the hoisting. The number of cars in each trip is shown by vertical components of the graph. For instance, at 7:57 the first trip of 12 cars was landed at the locomotive cut-off point. As shown by the number in the circle, the locomotive was standing still one minute before proceeding through the motor run. Letters *N* and *S* indicate the side of the mine from which the trips arrived. Occasionally trips arrived simultaneously from both sides of the mine, as at 11:19 A.M.

On the day when this time study was carried out 61 trips came to the shaft bottom. The 24 trips with a total of 357 cars from the *N* side were delayed 1 hour 19 minutes, and the 37 trips with 878 cars from the *S* side were delayed 3 hours 24 minutes; that is, the loco-

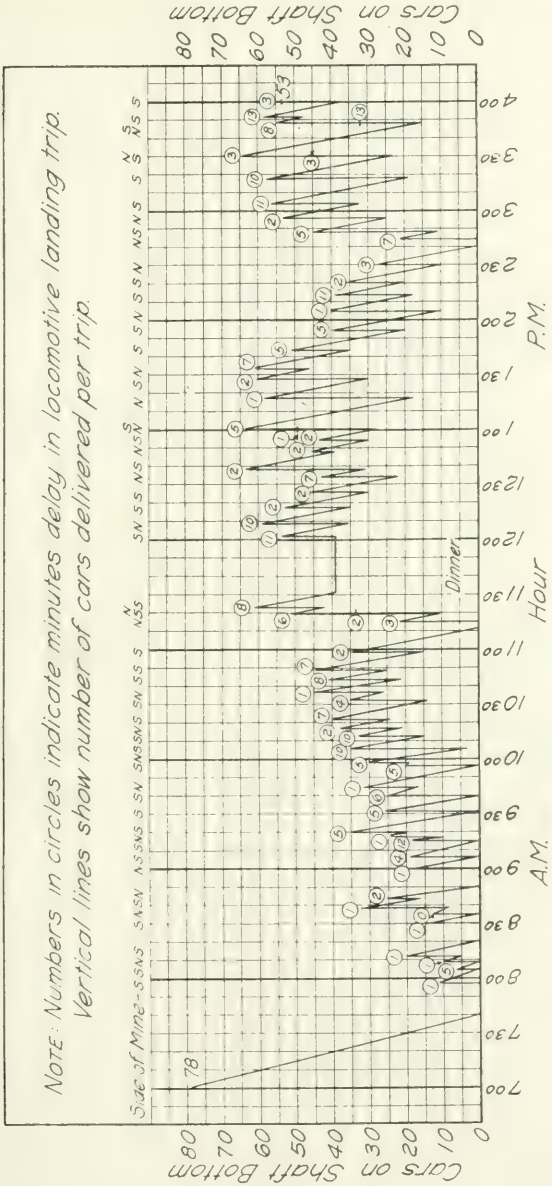

FIG. 19. GRAPH OF SHAFT-BOTTOM OPERATIONS

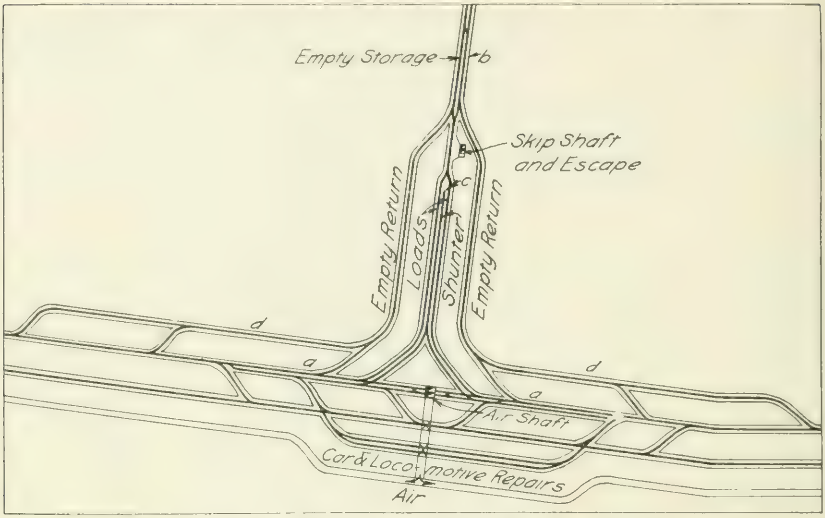

FIG. 20. MAP OF SHAFT BOTTOM FOR SKIP HOISTING

motives waited a total of 4 hours 43 minutes before proceeding from the cut-off point to the empty-storage track.

19. *Shaft Bottoms for Skip Hoisting.*—Prior to 1917 there were in Illinois only three installations at which skip hoists were used. Two of these were at small-capacity mines and the end-gate type of car was used; the third mine had an average daily production of between three and four thousand tons and a bottom-dump car was used.

Since 1918 there have been opened several large shaft mines in which skips, rotary dumps, and solid-end cars are installed. The capacity of these skips is between 10 and 12 tons.* At one of the mines noted, a trial record of 1000 tons in one hour was obtained in 1920. The rotary dump permits the use of the solid-end car, thus giving a more rigid construction, one of the greatest sources of trouble in mine-car construction being the loose end-gate; it also simplifies track layout as the car may be run in either direction. Thus, at one mine the track layout is such that the position of a car on alternate trips from the shaft bottom to the face is reversed, which could not be the case with the ordinary self-dumping cage layout.

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* A detailed discussion of skip hoisting will be found in an article by Allen and Garcia in the Trans. Am. Inst. Min. & Met. Engr. for 1921, reprinted in "Coal Age," March 17 and 24, 1921.

The average shaft bottom arrangements for several mines at which skips are used are shown in Fig. 20. As all these mines are still in the development stage, costs per ton for handling coal on the bottom are not yet available.

The locomotives coming to the shaft bottom are detached from their trips at points *a*, passing thence into the empty-return entries, while the loaded cars move toward the main shaft under control of a pusher locomotive or shunter traveling on the auxiliary track *c*. All loads pass the hoisting shaft over a single track and through a rotary dump. The empties return from *b* through the return entries to be picked up by the locomotives and hauled back to the workings through entries *d*. Double trackage in *b* permits a continuous influx of empty cars without interruptions due to outgoing empty trips. The auxiliary or air shaft is conveniently located on the main haulage entry for cage hoisting when necessary. Fig. 21 shows in vertical cross-section a typical skip-hoisting equipment, including rotary dumps, storage hopper, and automatic measuring hopper.

The shaft bottom at one skip mine includes a rotary car-dumper and skip hoist for a capacity of 7000 to 8000 tons daily. The mine is divided into four sections, northeast, northwest, southeast, and southwest. On the west side of the shaft the locomotives cut off, pass through a run-around, obtain their empty trips in the back entry, and then proceed westward along the back entry to the main west haulage roads. From the east side of the mine the loaded trips are pulled past the shaft along the back entries to the west main shaft approach. The locomotives are detached at the same point as are those from the west, then pass through the run-around to the back entry where they obtain their empty trips and proceed directly to the northeast or southeast portions of the mine. Loaded cars are handled singly by a mechanical car-haul to the weigh scale. A special track is provided for switching broken cars. All material is handled at the auxiliary hoist shaft located to the west of the main shaft. The installation is made complete by a motor-generator room and necessary repair and supply shops near the auxiliary hoist shaft.

At another mine the shaft-bottom arrangement includes a skip hoist for the coal and an auxiliary air-and-materials shaft provided with cages. The main-line haulage locomotives cut off in the main

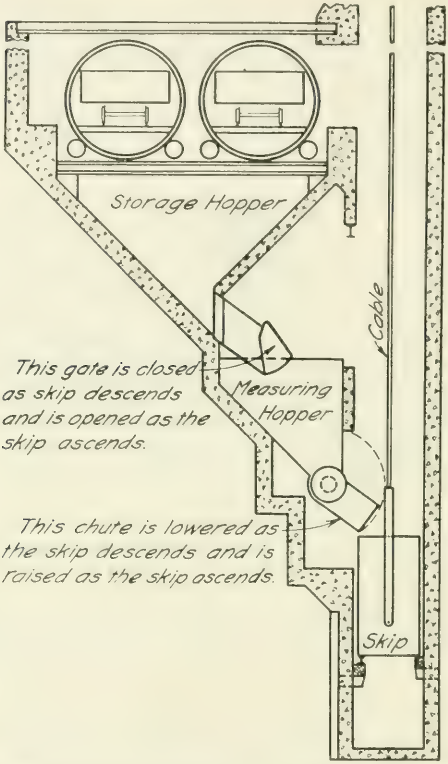
Storage Hopper
Cable
This gate is closed as skip descends and is opened as the skip ascends.
Measuring Hopper
This chute is lowered as the skip descends and is raised as the skip ascends.
Skip

FIG. 21. VERTICAL CROSS SECTION—SKIP-HOISTING SHAFT

entry and go through the empty run-around to the empty-storage track back of the shaft. A pusher locomotive pushes the loaded trip toward the shaft as described in connection with Mine A, page 56.

The empty side of the shaft beyond the skip-pit is provided with two tracks so that the empty cars can continue to pass to the storage tracks, even though the locomotive may be pulling out an empty trip. The auxiliary shaft is so located that cars may be conveniently sent from the main haulage to the cages or returned from the cages to the main haulage.

The arrangement at still another skip mine provides for the loaded cars to be detached along the main haulage road at a point where the locomotive enters the cross-cut leading to the empty run-around, and the loaded trip continues to the dump by gravity, assisted by three car-hauls. These hauls are electrically driven and operated by one dump-man located near the shaft. On the empty-storage side the locomotive is coupled to the end of the trip and, after the cars have passed through the car-dump, pulls it toward a switch, then pushes it through one of the cross-cuts at the right or left to the empty tracks where the main-line locomotives receive their trips. Instead of using car-hauls on the loaded side of the shaft and a locomotive on the empty side, the same arrangement of tracks can be used and the loads carried by gravity to the dump, while the locomotive may pass through the run-around and take the empty trip from the tracks back of the shaft. An arrangement of tracks at the auxiliary shaft is such that the coal may be caged from either side of the shaft and the empty cars returned to the same side from which they were caged.

IV. Main Line and Gathering Haulage

20. *General Considerations.*—Main-line haulage means that portion of the haulage system between the shaft bottom and the gathering partings where the cars are collected from the rooms and made into trips.

The questions to be considered in connection with the main haulage are, therefore, supply of ample power, condition and grade of track, kind and condition of equipment (such as locomotives and cars), speed of travel, suitable and properly-spaced turnouts or pass partings when single track is used, maintenance of a schedule of trips that will cause a minimum of delay at the terminal point and at the pass partings, and prevention of accidents.

The data for the mines studied show that the time spent by the locomotives on the main line is generally less than that consumed in making up trips on the partings, delivering loaded trips on the bottom, and picking up empty trips on the bottom for return to the parting, providing the main-line haulage distance is not more than one mile. The workings from which each locomotive receives the cars should be concentrated so that the locomotive does not have to go to widely separated gathering partings. There should also be an adequate reserve of empty cars on the shaft bottom so that the incoming locomotives are not required to wait for their return trips.

Satisfactory performance on the main haulage is not so much a factor of speed of running as it is of continuous and regular operation. If the haulage system is properly laid out and operated, a high speed of haulage is unnecessary. A slow, uniform speed gives increased safety to employees, both to those engaged in haulage and to others who may be compelled to use the haulage roads. A conservative speed results also in less spillage of coal along the track, less raising of dust, and less cost for repairs to equipment. Usually a maximum speed of six to eight miles per hour can be adopted with increased safety. Only occasionally need the trips be run at higher speeds, as when making up time lost through unusual or irregular causes.

The trolley type of locomotive is generally used for main-line haulage and is fairly well standardized for the conditions that exist in Illinois mines. The locomotives vary from 6 to 20 tons in weight, many of the recent installations being of the 15-ton type. In the larger mines as the length of main-line haul increases the size of the locomotive used for this duty also increases; because generally the greater the capacity per locomotive the smaller number required for a given tonnage, provided the haulage layout is properly designed and there are ample side-tracks. During a shift of eight hours and under suitable operating conditions, a 15-ton locomotive should easily haul on the main line from 1500 to 2000 tons of coal a distance of one mile, but as shown in Table 4 this is being done in very few of the mines studied.

Generally the main-line haulage and gathering are kept separate. At a few mines, however, the locomotives used on the main haulage also gather from the faces and thus run directly from the faces to the shaft bottom with comparatively small trips of cars.

21. *Location of Partings.*—In connection with the gathering of the cars from the rooms, the location of the partings with respect to the room entries materially affects the efficiency of both gathering and main haulage.

It is important that the work of gathering be concentrated so as to reduce the number of partings, the number of cars required, and the distance that either mules or locomotives must travel in unproductive work. The partings should be so advanced that they will always be within a certain standard distance of the working face. This distance varies widely in different mines, but for mule haulage it is generally about 800 to 1200 feet and for locomotives 800 to 2000 feet. In some mines the partings are placed centrally with respect to four panels, the cars being back-hauled from two panels to the parting. The disadvantage of back-hauling from the older panels, in which there may be only a few rooms working, may be more than compensated for by the advantages given to newly developed territory from which the bulk of the hauling is done. Thus each time a parting is moved, whether it be after intervals of two, three or more years, the point should be selected to insure the greatest return for the expense involved; that is, it should be as close as possible to the ''center of production'' of the tonnage to be produced during a given installation period. Partings are made either by widening a single track entry so that a double track may be installed or by driving an extra passage in

TABLE 4

MAIN LINE HAULAGE IN EIGHT TYPICAL LARGE ILLINOIS COAL MINES

Mine	Ave. Tons Daily	Wt. Car in Tons	Tons Coal per Car	Locomotives			Distances Hauled, Parting to Bottom			Cars per Trip			Cars per Day Parting to Bottom		
				No	Wt.	Kind	Max.	Min.	Ave.	Max.	Min.	Ave.	Max	Min.	Ave
B	5200	2.35	4.30	7	15-ton	Trolley	6400	2600	4575	21	7	17	312	70	155
C	4500	2.00	5.00	1 1 1	6-ton 10-ton 15-ton	Tandem Reel and Trolley Reel and Trolley	4660	2710	3670	16	8	13	80	40	70
D	5000	1.20	3.50	1 3	12-ton 15-ton	Trolley	6900	4000	5164	23	10	15	200	40	107
E	3800	1.30	2.30	2 2	10-ton 13-ton	Trolley	8350	4350	6217	27	18	23	380	110	277
F	3600	2.00	4.50	2	15-ton	Trolley	4800	2400	3422	20	10	16	120	40	84
G	2600	1.85	3.30	2	13-ton	Trolley	5200	2000	3867	25	10	18	220	20	137
H	3400	1.60	3.25	1 1	12-ton 13-ton	Trolley	4000	2000	3440	30	20	24	300	80	195
I	4000	1.39	3.00	2 2	12-ton 15-ton	Trolley	8000	4900	6087	26	24	25	235	90	166

TABLE 4 (CONTINUED)

MAIN LINE HAULAGE IN EIGHT TYPICAL LARGE ILLINOIS COAL MINES

Mine	Tons Coal per Day Parting to Bottom			Tons Coal per Day per Locomotive			Ton-Miles Coal per Locomotive			Total Ton-Miles per Locomotive			Loco.-Miles per Day		
	Max.	Min.	Ave.	Max.	Min.	Ave.	Max.	Min.	Ave.	Max.	Min.	Ave.	Max	Min.	Ave.
B	1350	300	655	1350	300	749	1704	150	788	3386	290	1518	24	9	17.4
C	400	200	350	1925	1105	1502	1121	688	902	2018	1238	1624	35	21	29.0
D	700	140	375	1320	1300	1314	1381	1081	1201	2411	1837	2047	57	38	48.7
E	874	253	637	1127	805	958	1380	920	1162	2950	1960	2479	44	35	40.8
F	560	160	370	1750	1580	1665	1233	912	1073	2330	1724	2028	34	25	29.5
G	735	70	450	1390	1310	1350	1151	620	886	2441	1315	1879	40	18	29.0
H	975	260	634	1805	1365	1585	1151	930	1040	2284	1846	2065	26.5	25	25.5
I	715	270	500	1100	840	1000	1521	890	1146	2931	1715	2209	39	25	30.3

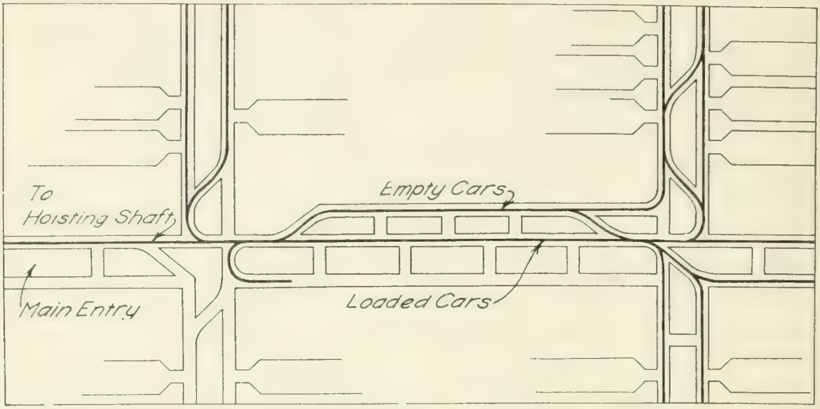

FIG. 22. TYPICAL PLAN OF MINE PARTINGS

the pillar as shown in Fig. 22, which shows also a diagonal arrangement at the entry crossings. Fig. 23 shows the diagonal connection between entries at a prominent mine.

22. *Procedure of Gathering.*—The methods of distributing cars to and gathering them from the rooms vary with the method of working, the agreement between operators and miners, the track arrangement, and the weight of the car. In Illinois, although each miner is assigned a definite room, two men usually load together in one room while an adjoining room is being undercut, so that on any day, even if the entire working force of miners is busy, coal will be loaded in only half the total number of rooms. Unless the car is too heavy or the grade conditions unfavorable, the miners usually push the empty car to the face but the loaded car is always taken from the face by a locomotive or a mule.

Gathering by Locomotive

The procedure in gathering by locomotive usually conforms to one of the three methods, illustrated in Fig. 24 for a panel of 14 rooms on each entry:

1. The empties are left at the room necks but the locomotive goes to the room face for each loaded car. There are two variations of this general method, (a) and (b).

(a) The locomotive pushes the empty trip into a given panel, and distributes the empty cars to and gathers the loaded cars from seven rooms, as numbers 1, 3, 5, 7, 9, 11 and 13, assumed to be working on a given day. Assuming that the miners from rooms 1 and 2 are loading in room 1, an empty car is detached from the inbye end of the trip and delivered into the switch for room 2 which has been or is being cut by the machine men. Proceeding inbye, cars are switched into rooms 4, 6, 8, etc. to room 14. The locomotive takes the loaded car from face of room 13 as far as the switch for room 11, where it is detached from the locomotive. At this time the condition is as shown in Fig. 24*a*. The locomotive next takes the loaded car from room 11. This car is coupled to the car from room 13 and the locomotive proceeds outbye, similarly taking the loaded cars from rooms 9, 7, etc. When the loaded car has been taken from room 1, the trip contains seven cars which are then hauled to the parting. The miners then push the empty cars to the working faces.

(b) The locomotive collects the loaded cars as in (a) and takes the loaded trip to the parting; then, returning with an empty trip, leaves the empty cars in the necks of the rooms that are being worked,

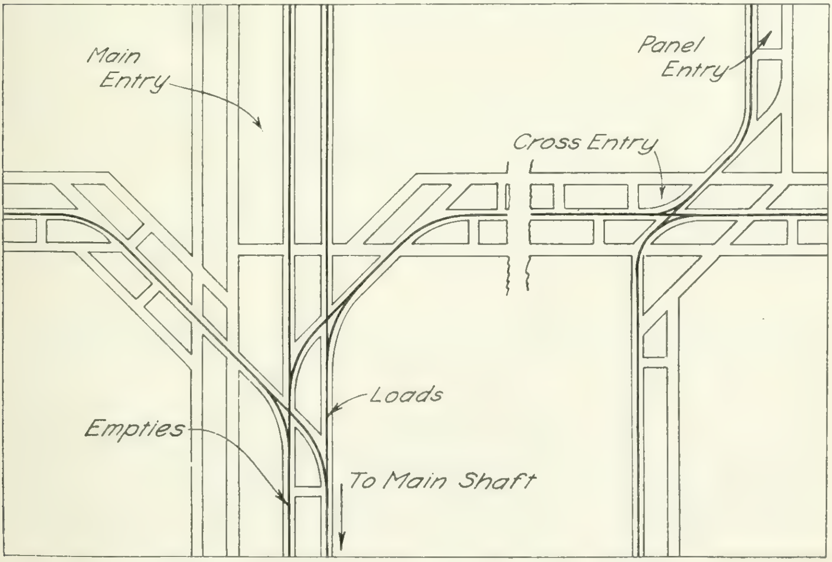

FIG. 23. DIAGONAL CONNECTIONS BETWEEN ENTRIES

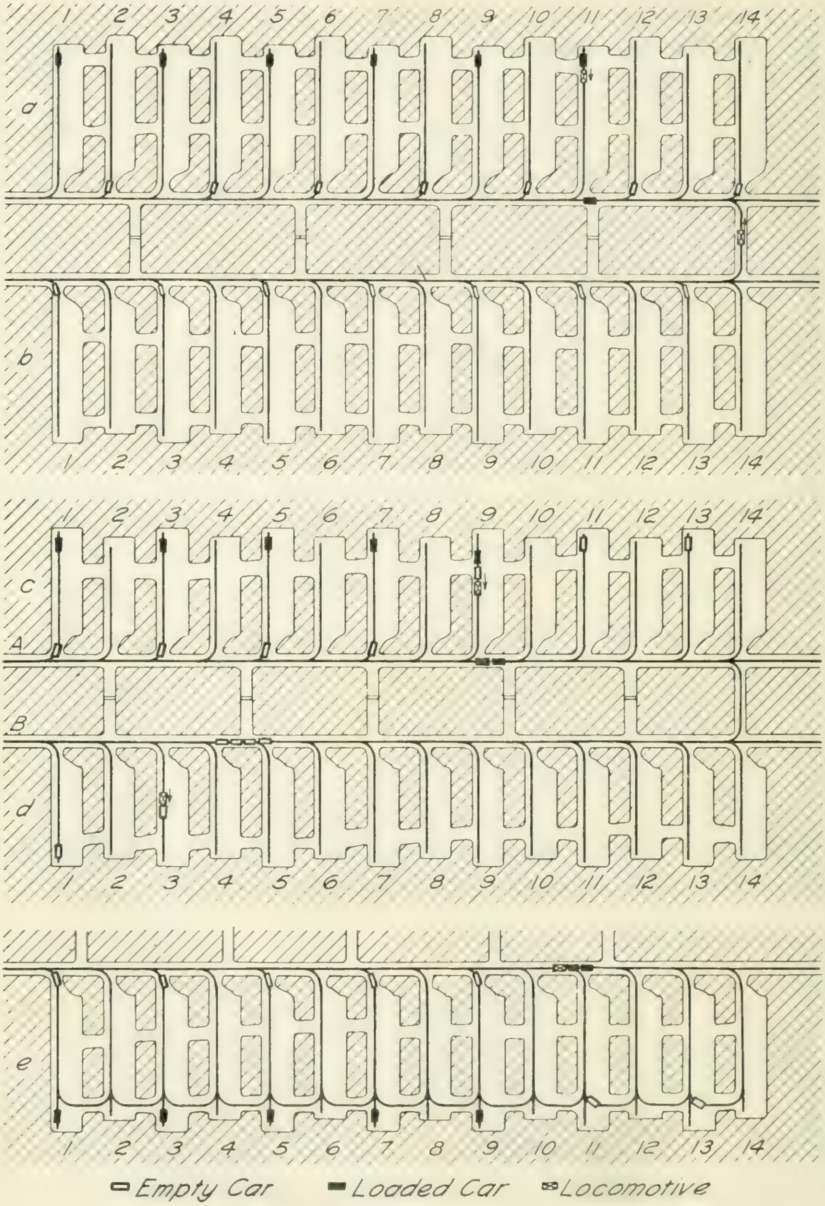

FIG. 24. METHODS OF GATHERING BY LOCOMOTIVES

as rooms 1, 3, 5, etc. After leaving the last empty car in room 13 the locomotive goes through the last open cross-cut to the parallel entry, as shown in Fig. 24*b*, and collects the loaded cars on its way out; or if this is not practicable it proceeds without any cars to another entry to make up a loaded trip.

2. Cars are taken to and from the working face by the locomotive by either of two systems, (c) or (d).

(c) An empty car is cut off at each of the rooms 1, 3, 5, to 13 as the locomotive proceeds along the entry. When the empty trip has thus been distributed and room 13 is reached, the empty car is pushed into the face of room 13 and coupled to the loaded car which is then pulled out and pushed along the entry to a point just inbye of the switch into 13, and there blocked and uncoupled from the empty car. The locomotive then returns into room 13 with the empty car which is pushed to the working face and there left. The locomotive returns to the loaded car at the mouth of room 13 and takes it to the switch just inbye of room 11, where the locomotive is uncoupled. The empty that has been left at the mouth of room 11 is then pushed up to the face and coupled to the loaded car. This procedure is repeated at each of rooms 9, 7, etc. After an empty car has thus been placed at the face of each room the trip of 7 loaded cars is taken to the parting. Fig. 24*c* shows the condition along the entry after the locomotive has gathered two loaded cars from rooms 13 and 11 and is pulling the loaded car out of room 9.

(d) The loaded cars are taken successively from the faces of rooms 13, 11, 9, to 1, and the loaded trip of 7 cars is hauled out to the parting. Returning, the locomotive pushes the empty trip past the switch of room 1, where it is blocked. Then the car next to the locomotive is uncoupled from the empty trip and is pushed by the locomotive to the room face. Similarly, the locomotive pushes an empty up to the face of each of rooms 3, 5, to 13. Fig. 24*d* shows the locomotive pushing an empty car into room 3. The locomotive then proceeds to the parallel entry through a cross-cut at the face; otherwise it backs along the same entry to the main entry and thence to another gathering section. If track is maintained in both of the room entries *A* and *B* it saves time of the locomotive to connect these entries with a track through the last cross-cut. This method requires the loaders to wait while the locomotive goes to the parting with a loaded trip and returns with an empty trip.

3. The empty cars are taken to a room cross-cut switch near the face by a locomotive or mules. In this way the distance that a car is pushed by the miner is decreased. The method is therefore intermediate between methods 1 and 2 in the amount of hand-pushing of the cars, and is particularly applicable also when the cross-cut is worked more as a separate room or place than as an ordinary narrow cross-cut. The empty cars are placed in order just inside the room necks, from room 1 to room 13, as the locomotive proceeds toward the face of the entry. At room 13 an empty is pushed up into the room and placed on the cross-cut switch. The loaded car is then taken from the face to a point on the entry just inbye the switch to room 11. The empty in the neck of room 11 is then pushed to the room cross-cut switch and the loaded car at the face of room 11 is brought to the entry and coupled to the car from room 13, as shown in Fig. 24*c*.

If a room cross-cut is being driven wide so that it is practically a room, and its loaded car is in the cross-cut, the empty will be left just beyond the cross-cut switch along the room track while the locomotive goes into the cross-cut for the loaded car. If the loaded car is at the face of the room the empty is placed on the cross-cut switch and the loaded car then taken from the face of the room.

With this method track need be maintained in one room only, while two or more rooms on each side are served by spur tracks through the last open cross-cuts. In this practice the locomotive enters room 1 with two empty cars which are switched at the face and the loads are then taken out by the locomotive. Extra switchlaying is required each time the room is advanced a cross-cut length, but this outlay is compensated for by the saving of both time and upkeep in room haulage and equipment. Moreover, there is some advantage in a shorter travel of the mining machine in these rooms which, being kept off the entry haulage roads to a greater extent, interferes less with haulage.

Gathering by Mules

For hauling cars in rooms and for short entry hauls mules are effectively used, hence the parting should be kept as close to the working face as practicable. The procedure is very similar to that followed in gathering by locomotives except that smaller trips are necessary. Usually from two to three cars only are hauled to the parting while, at times, on account of adverse grades, a mule can pull only one loaded car. When the driver delivers the empty cars to the room faces he does so on the return trip after the loads have been taken to the parting.

In some mines the loaded cars are taken from the room faces to the entry by mules and there made into trips by the gathering locomotive and taken to the parting. While the locomotive is returning with the empty cars the same driver and mule are employed in gathering a similar trip of cars from the rooms in the adjoining entry. In delivering the empty cars the locomotive pulls the cars into the room entry where the trip rider cuts off one car from the rear of the trip at each working room. If a track has been laid through the last open cross-cut between two panel entries, the locomotive, after delivering all of the empties in one entry, can pass through the cross-cut to the adjoining entry, gather the cars in that entry into a trip, and take them to the parting.

23. *Performance of Main-Line Locomotives.*—In order to secure comparative data in regard to performance at different mines operating under different methods of main-line and gathering haulage, a study was made of the number of cars, the weight of coal, and the distances hauled, both in gathering and on main haulage; thus data were obtained for ton-miles per day per locomotive, which is the measure of performance used for comparing the operation of locomotives on standard surface roads.

Table 4 gives a summary of performance data for main-line locomotives at the mines listed as B to I, inclusively, under the shaft bottom discussion, pages 58 to 63. The mines covered by Table 4 are all large producers and have modern equipment. This table shows a considerable variation in the daily performance at different mines such as the average number of cars hauled per trip, the average number of cars hauled per day, and tonnage or cars per day per parting, but the real basis for comparison is the average ton-miles of coal hauled per locomotive and the average locomotive-miles per day. The detailed study of several of these mines shows a similar variation in the work performed by different locomotives in the same mine, suggesting that at many mines a re-adjustment of locomotive schedule might be made with advantage. With the exception of Mines B and E in each mine of this group the main-line locomotives average over 1000 tons of coal per day delivered to the shaft bottom. At Mine H two locomotives average 1585 tons per day with an average haul of 3440 feet. The total ton-mileage per locomotive varies between 290 and 3386; the two greatest averages, 2479 and 2209, being made at the two mines having the longest average hauls of 6217 and 6087 feet respectively. Of the two mines that lead in production one shows

TABLE 5
MAIN LINE HAULAGE FOR EIGHTEEN MINES
Group of Mines Producing between 1500 and 3000 Tons Daily

Mines	Ave. Daily Tonnage	No. Years in Oper.	Mine Car		Loco.		No. Partings	Ave. Dist. Haul. feet
			Empty Wt. tons	Coal Wt. tons	No.	Wt		
1	2200	15	1.25	3.20	3	12-ton	5	3500
2	2600	14	1.00	3.05	2	14-ton	8	4000
3	2500	9	1.50	4.25	2	12-ton	10	4025
4	1800	12	1.45	3.25	2	13-ton	7	2700
5	2250	15	1.43	2.25	3	10-ton	10	4400
6	2200	17	1.00	2.40	2	10-ton	7	5000
7	1800	17	0.75	2.00	2	10-ton	6	5000
8	2500	22	0.70	3.00	3	10-ton	7	3750
9	1700	15	*	2.75	2	12-ton	4	2700
10	2500	4	1.05	2.90	3	10-ton	8	1950
11	1770	20	*	2.20	3	12-ton	5	5280
12	2200	20	*	2.25	3	12-ton	5	6600
13	1500	15	*	2.00	2	13-ton	5	6000
14	2000	15	*	2.50	3	15-ton	6	7000
15	3000	20	1.50	3.00	4	13-ton	11	6000
16	1700	31	1.50	3.00	3	13-ton	7	4500
17	1800	15	1.15	2.70	3	10-ton	6	5100
18	1600	16	1.10	3.00	3	10-ton	6	3500
Ave.	2100	16	1.18	2.78	3	12-ton	6	4500

*Empty car weight not available. Total ton-miles per loco. approximately 1.81 times ton-miles coal.

the greatest average daily mileage per locomotive while the other shows the least. It is interesting to note that Mine B, having the greatest production, puts the least average duty in total ton-miles per day upon its locomotives, but at the same time puts both extremes of such duty upon them. This mine has also the fewest ton-miles of coal per locomotive per day.

TABLE 5 (CONTINUED)

MAIN-LINE HAULAGE FOR EIGHTEEN MINES

Group of Mines Producing between 1500 and 3000 Tons Daily

Mines	Ave. Cars per Trip	Ave. Trips per Day	Ave. Cars per Loco. per Day	Ave. Tons Coal per Loco.	Ave. Ton-Miles Coal per Loco.	Ave. Ton-Miles per Loco.	Ave. Loco. Miles per Day
1	15	16	240	700	465	750	20
2	18	24	430	1300	1000	1670	36
3	20	15	300	1250	950	1620	23
4	20	15	300	900	460	690	15
5	20	17	340	750	600	1350	28
6	23	20	460	1100	1000	1830	38
7	22	20	440	900	850	1500	38
8	18	16	280	840	600	900	23
9	18	16	280	850	440	*	16
10	12	24	280	840	310	510	18
11	16	17	270	600	600	*	34
12	20	16	320	700	900	*	40
13	16	20	320	650	740	*	45
14	17	16	270	670	900	*	42
15	15	16	250	750	850	1700	36
16	12	16	190	570	490	980	27
17	22	10	220	600	580	1080	20
18	10	18	180	540	360	620	24
Ave.	17	17	300	800	670	1320	29

*Empty car weight not available. Total ton-miles per loco. approximately 1.81 times ton-miles coal.

TABLE 6

PERFORMANCE OF FIVE 15-TON MAIN-LINE LOCOMOTIVES IN A LARGE ILLINOIS MINE For One Shift

Locomotive	(a)		(b)		(c)	
No. Trips	14		13		15	
Ave. No. Cars per Empty Trip	18		18		18	
Ave. No. Cars per Loaded Trip	19		18		18	
Total Loads	275		242		292	
Total Tons Coal	1133		997		1203	
Ave. Distance Hauled	4850		3700		4950	
Ton-Miles Coal	1040		700		1130	
Locomotive Miles	26		18		28	
Analysis of Time	**Min.**	**Per Cent**	**Min.**	**Per Cent**	**Min.**	**Per Cent**
Running Time on Main Line	197	43	197	43	198	44
Loaded	102		94		105	
Empty	95		103		93	
Switching Time	102	22	87	19	92	20
Motor Run	24		22		28	
Empty Run Around	27		20		23	
Inside Parting	51		45		41	
Total Running Time	299	65	284	62	290	64
Total Delays	161	35	177		163	
Delays at Shaft Bottom						
Blocked by Loads	87		79		52	
Waits for Empties	15		19		21	
Delays, Inside Partings						
Waits for Loads	59		36		90	
Repairs	..		43		..	
Total Operating Time	460	100	461	100	453	100

TABLE 6 (CONTINUED)

PERFORMANCE OF FIVE 15-TON MAIN-LINE LOCOMOTIVES IN A LARGE ILLINOIS MINE
For One Shift

Locomotive	(d)	Ave.	(e)*
No. Trips	14	14	14
Ave. No. Cars per Empty Trip	23	19	16
Ave. No. Cars per Loaded Trip	23	19	15
Total Loads	543	288	208
Total Tons Coal	1413	1186	857
Ave. Distance Hauled	4750	4562	1500
Ton-Miles Coal	1270	1035	240
Locomotive Miles	25	24	8

Analysis of Time	Min.	Per Cent	Min.	Per Cent	Min.	Per Cent
Running Time on Main Line	189	44	195	43	118	26
Loaded	98		100		68	
Empty	91		95		50	
Switching Time	92	21	93	21	55	12
Motor Run	34		27		....	
Empty Run Around	24		24		46	
Inside Parting	34		43		9	
Total Running Time	281	65	288	64	173	38
Total Delays	150		163		278	
Delays at Shaft Bottom						
Blocked by Loads	90		77			
Waits for Empties	23		20		201	
Delays, Inside Partings						
Waits for Loads	37		56		77	
Repairs	..		10			
Total Operating Time	431	100	451	100	451	100

* Locomotive (e) does relay duty.

TABLE 7

GATHERING HAULAGE IN EIGHT TYPICAL LARGE ILLINOIS COAL MINES

Mine	Ave. Tons Daily	Locomotive			No. Mules	No. Partings	Approx. Dist. Hauled to Partings			Cars per Trip to Partings			Cars per Day to Partings		
		No.	Wt.	Kind			Max	Min.	Ave.	Max.	Min.	Ave.	Max.	Min.	Ave.
B[1]	5200	12 11	8-ton 6-ton	Reel and Trolley	0	8	2200	600	1137	6	6	6	190	85	146
C	4500	12	8-ton	Storage Battery	0	12	1900	800	1000	8	2	7	80	35	69
D	5000	9 6	6-ton 8-ton	Reel and Stor. Bat.	5	14	2100	800	1366	14	2	9	200	40	107
E	3800	0	.	....	17	4	1300	700	1055	3	2	2	375	180	294
F	3600	16	6-ton	Reel and Trolley	0	9	2000	800	1389	15	11	14	126	42	86
G	2600	14	6-ton	Reel and Trolley	0	6	1700	700	1036	8	2	5	225	20	138
H	3400	14	.	Storage Battery	0	6	1800	900	1150	10	5	8	280	80	176
I[2]	4000	14	6-ton	Dry Battery	3	8	2000	L.1100	L. 1379 M. 700	8	L. 8	L. 8 M. 1	240	L. 96	L. 178 M. 90

[1] In this mine one locomotive gathers to the shaft bottom. Its performance is included in Table 10, Main-Line Haulage.

[2] Locomotives gather exclusively to seven partings—mules exclusively to one parting.

TABLE 7 (CONTINUED)

GATHERING HAULAGE IN EIGHT TYPICAL LARGE ILLINOIS COAL MINES

Mine	Tons Coal per Day to Partings			Tons Coal per Loco. or Mule			Ton-Miles, Coal per Loco. or Mule			Total Ton-Miles, per Loco. or Mule			Total Daily Mileage per Loco. or Mule			No. Miners per Parting			No. Machines per Parting		
	Max.	Min.	Ave.	Max.	Min.	Ave.	Max.	Min.	Ave.	Max.	Min.	Ave.	Max.	Min.	Ave.	Max.	Min.	Ave.	Max.	Min.	Ave.
B[1]	820	360	617	300	170	228	74	26	45	141	50	86	5.6	2.1	3.6	71	29	57	7	2	5
C	400	175	327	400	175	340	144	30	66	259	54	119	7.2	1.5	3.9	34	16	27	3	1	2
D	700	140	375	L. 350 M.180	225 140	300 150	L. 139 M. 36	48 21	85 29	L. 235 M. 60	80 36	144 49	L. 6.4 M.10.9	3.0 6.0	4.4 8.4	50	10	28	..	..	..
E	870	410	675	230	90	159	48	20	31	102	43	66	14.8	7.6	11.3	121	55	91	5	3	4
F	570	189	383	270	189	215	73	38	54	138	72	102	10.6	4.5	6.9	51	19	36	5	2	3
G	742	66	455	330	66	196	58	14	37	123	29	78	7.1	3.1	4.2	66	8	38	4	1	3
H	910	260	572	325	87	245	97	25	54	193	49	107	7.5	3.0	4.3	93	24	56	6	3	5
I[2]	720	L. 288	L. 535 M. 270	288	240	L. 267 M. 90	98	L.55	L. 70 M.12	189	L.106	L.135 M.23	8.2	L.4.6	L. 5.8 M.8.0	60	24	43	5	2	4

[1] In this mine one locomotive gathers to the shaft bottom. Its performance is included in Table 10, Main-Line Haulage.

[2] Locomotives gather exclusively to seven partings—mules exclusively to one parting.

TABLE 8

GATHERING HAULAGE IN SEVENTEEN TYPICAL ILLINOIS COAL MINES

Mine No.	Ave. Daily Tonnage	No. Men		No. Machines	No. Mules	Locomotives			No. Partings
		in Entries	in Rooms			No.	Wt.	Kind	
1	2200	32	178	21	0	11	6-ton	Reel	5
2	2600	26	154	17	0	13	6-ton	Reel	8
3	2500	70	170	22	0	14	6-ton	Reel	10
4	1800	58	130	17	0	8	6-ton	Reel	7
5	2250	66	217	15	25	. .	. . . . .	. . .	10
6	2200	40	164	17	. . 5	8 . .	5-ton	St. Bat.	7
7	1800	40	164	17	. . 2	8 . .	6-ton . . .	St. Bat.	6
8	2500	50	190	18	 17	1 3 . .	6-ton 6-ton	St. Bat. Reel	7
9	1700	30	113	11	12	. .	. . . . .	. . . .	4
10	2500	54	349	. .	0	14	5-ton	St. Bat.	8
11	1770	49	132	18	24	. .	. . . . .	.	6
12	2200	11	177	22	28 1	. . . .	. . . .	. . . .	5 1
14	2000	. .	300	. .	. . 12	4 . .	5-ton	St. Bat.	8
15	3000	23	252	11	0	12	5-ton	St. Bat.	6
16	1700	16	200	2	. . 2	6 . .	5-ton	St. Bat. . . .	7
17	1800	46	125	20	. . . 4	2 5 . .	. . . .	St. Bat. Reel . . .	6
18	1600		. .		20		. . .	.	6

TABLE 8 (CONTINUED)

GATHERING HAULAGE IN SEVENTEEN TYPICAL ILLINOIS COAL MINES

	Mine No.	Ave. Dist. Haul. feet	Ave. Car Trips	Ave. Trips per Day	Ave. Cars per Loco. or Mule	Ave. Tons Coal per Loco. or Mule	Ave. Wt. Tons Coal per Car	Ave. Ton-Mi. Coal per Loco. or Mule	Ave. Ton-Mi. per Loco. or Mule	Ave. Loco. or Mule-Mi. per Day
	1	L. 1500	6—	11	63	200	3.20	56.8	102.0	6.24
	2	L. 1100	8+	8	67	200	3.05	41.7	69.8	3.34
	3	L. 1650	5	8	40	170	4.25	53.1	90.6	5.00
	4	L. 850	7	10	70	225	3.25	36.2	68.8	3.22
	5	M. 800	1½	27	40	90	2.25	13.6	30.9	8.18
	6	L. 1000	6+	14	85	200	2.40	37.9	70.8	5.30
		M. 500	3	15	45	110	2.40	10.4	18.9	2.84
	7	L. 1000	7+	14	100	200	2.00	37.9	66.4	5.30
		M. 500	3	16	48	100	2.00	9.5	16.3	3.03
	8	L. 900	7	10	70	200	3.00	34.1	52.5	3.41
		M. 900	1–2	25	32	100	3.00	17.0	24.6	8.53
	9	M. 1000	3	17	50	140	2.75	26.5	49.7	6.44
	10	L. 1200	6+	10	62	180	2.90	40.8	70.4	4.54
	11	M. 1000	1	32	32	70	2.20	13.2	24.8	12.11
	12	M. 800	1	35	35	40	2.25	6.1	11.4	10.60
		M. 650	1	40	40	75	2.25	9.2	17.2	9.85
	14	L. 1200	7—	12	80	200	2.50	45.4	82.2	5.45
		M. 800	2	20	40	100	2.50	15.1	28.3	6.06
	15	L. 1100	6—	14	83	250	3.00	52.1	103.9	5.83
	16	L. 1000	6—	15	87	260	3.00	49.2	98.7	5.68
		M. 700	1	24	24	70	3.00	9.3	18.8	6.36
	17	L. 1800	7+	11	80	210	2.70	71.5	136.3	7.50
		M. 1300	1	32	32	85	2.70	20.9	39.0	15.75
	18	M. 900	1	27	27	80	3.00	13.6	23.7	9.20
Ave. Performance	109 Loco.	1227	..	11.05	71—	204.9	3.04	46.94	84.98	5.21
	54 Reel-trolley Loco.	1341	..	9.30	61	197.0	3.39	49.26	86.80	5.10
	55 St. Bat. Loco.	1115	..	12.76	80+	212.4	2.70	44.67	83.20	5.30
	152 Mules	869	..	27.36	36	82.5	2.18	13.60	25.30	9.05

Somewhat similar data are given in Table 5 for 18 mines having a variation in production from 1500 to 3000 tons per day. Most of these mines, being older than those listed in Table 4 and therefore having mine cars that are generally of less capacity, have lower ton-mileages.

Table 6 has been prepared from data taken in one of the largest coal mines of Illinois to show the average daily performance of the main-line locomotives. These data cover one full shift of eight hours. The average distance traveled by a locomotive per round trip was 1.72 miles. Each locomotive was on duty approximately 94 per cent of its full shift, and an analysis of its actual operating time is given. Thus locomotive (a) was on duty 460 minutes or during 96 per cent of its 8-hour shift, but of this time it actually operated only 299 minutes or 65 per cent of the 460 minutes. Four kinds of delays consumed 161 minutes of this locomotive's time and the average delay per locomotive per shift was practically 2.72 hours.

24. *Performance of Gathering Locomotives.*—Table 7 covers the data on gathering haulage for the same mines as, and in a manner similar to, Table 4 for main haulage. Table 8 similarly covers seventeen of the mines in Table 5. Owing to the constantly changing distances that cars are hauled in gathering from the same territory, it was impossible in the time available to obtain accurate data for each car moved during the period when the study was made in each mine, but a distance from a central point in the panel to the parting was assumed as the average travel for the cars gathered from the given panel, and the average weight of coal per car was also assumed for the mine during the given period. While these assumptions may not give exact results for any given day, they probably represent the average operating conditions of any given mine and are of value in comparing the performance of locomotives in different mines and in different sections of the same mine.

Gathering is performed by locomotives exclusively in five of the selected eight large Illinois coal mines, by mules exclusively in one of these mines, and by both locomotives and mules in two mines. These eight mines utilize for gathering haulage 108 locomotives and 25 mules.

Improvements in reel and crab locomotives permit their use in even the most difficult working places.

The average distance covered by main-line haulage in these eight typical mines is 4555 feet, while the average distance traveled in gathering in these same mines is but 1141 feet, or approximately one-fourth the main haulage travel. The sizes of trips in the two stages of haulage are as 7 cars in gathering to 19 cars in main haulage. In main-line haulage a locomotive averages 31.3 miles per day, whereas in gathering it averages but 4.8 miles. In total ton-mileage per locomotive the figures for the two classes of haulage are as 111 in gathering to 1985 in main haulage, or about as 1 to 18.

As the activities of coal mining continually alter underground workings, the tables must be accepted as statistically accurate for a relatively short period only, and only for the dates upon which the data were secured. The method of diagramming and listing the data for each of the mines is illustrated by giving the diagrams and tables for Mine A, having one-stage haulage, and for Mine D, having two-stage haulage.

25. *Details of Haulage Performance in Typical Illinois Mines.*—The detailed methods of representing the workings diagrammatically and of tabulating the haulage data used in compiling Tables 4 and 7 are given for two mines only—in Table 9 for Mine A, and in Tables 10 and 11 for Mine D. In Mine A, cars are hauled directly from the rooms to the shaft bottom by one set of locomotives, while in Mine D there is a distinction as to gathering and main-line haulage.

The following data regarding the handling of cars from the face to the shaft bottom will supplement the shaft-bottom data given in connection with Table 3.

Mine A

The daily production is 4500 tons. Each mine car weighs 2780 pounds empty, and holds four tons of coal, and there are 474 cars in the mine. Approximately equal amounts of coal reach the shaft bottom from the north and south sections of the mine, and the cars are brought directly from the working face to the shaft bottom by the same locomotives that gather the coal and operate on the main line. All empties are similarly hauled from the shaft bottom directly to the workings. It is believed that the system of gathering directly to the shaft bottom involves fewer delays.

Although the shaft was sunk at the approximate center of the original property the later development of the mine has been such that

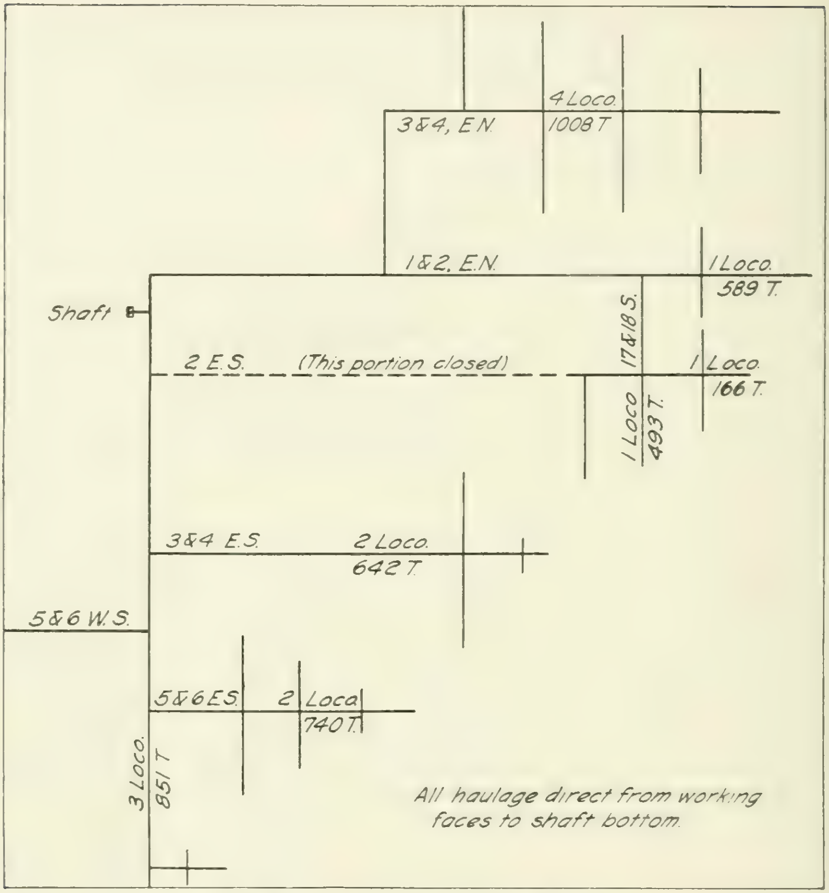

FIG. 25. HAULAGE DIAGRAM—MINE A

at present the shaft is near one side of the operating portion of the mine thus giving a rate of advance for the haulage roads about double that for a centrally located shaft. Fig. 25 is a diagrammatic sketch of the haulage roads in Mine A.

The main-haulage road is double track for 2500 feet in each direction from the shaft, north and south, so that there is no interference of incoming and outgoing trips. Adequate pass-partings also permit trips to pass conveniently in the cross-entries. Switch throwers

TABLE 9

HAULAGE—MINE A

Territories	Locomotives		Approx. Dist. Hauled	No. Cars per Day	Tons Coal per Day	Ton-Miles Coal per Loco.	Total Ton-Miles per Loco.	Daily Mileage per Loco.	No. of Men in	
	No.	Kind							Entries	Rooms
3 and 4, ES....	2	Reel and Trolley	6230	60	242.98	387.20	690.85	19.39	6	52
			7000	21	83.30					
			6300	79	316.20					
5 and 6, ES	2	Reel and Trolley	5550	78	313.90	403.74	720.36	21.88	10	48
			6600	33	123.45					
			5640	75	302.60					
Main S, & 5 and 6, WS...	3	Reel and Trolley	6700	25	95.30	307.21	548.13	15.51	18	55
			5000	74	300.10					
			5300	74	298.85					
			7300	40	156.60					
2 ES.......	1	Reel and Trolley	8230	49	195.00	409.05	729.83	24.61	4	15
			7800	18	71.15					
17 & 18 S.-2 ES	1	Reel and Trolley	7000	123	492.80	653.33	1165.68	37.12	0	38
1 and 2, EN	1	Reel and Trolley	6900	77	310.35	769.95	1373.75	41.84	0	28
			6825	61	243.05				0	28
			7500	9	35.35				4	0
3 and 4, EN	1	Reel and Trolley	7075	88	366.80	353.85	631.34	20.57	0	25
			7760	41	166.50				0	13
			7900	17	67.30				10	0
			7375	85	340.65				0	27
			8150	17	66.55				8	0
Averages	....		6707	..	....	410.80	732.95	22.50	..	..

are stationed at main junction points and extra flagmen are placed as required. The locomotives in this mine are combined trolley and storage-battery locomotives, and, while the greater part of the haulage is done by using current from a trolley wire, current from the batteries enables the locomotives to reach the working faces. This system is here held preferable to the use of reel locomotives in that it reduces delay in changing from trolley service and lessens the peak loads on the power circuit. Each locomotive averages daily seven trips of from 6 to 14 cars each. The grades in the room are nearly level and about 95 per cent of the empty cars are left by the locomotives at the room necks, but the loaded cars are gathered from the faces by the locomotives. The average haul of a locomotive is 6707 feet.

Mine A has the same daily production as Mine C in which the haulage is divided into two stages. For Mine A the average total ton-miles per locomotive is 733. In Mine C the average total ton-miles per locomotive per day in gathering is 119 and in main haulage is 1624, and the general average for all locomotives is 420. Similarly, the respective total of ton-miles for gathering haulage and main haulage in Mine D (which has a daily production of 5000 tons) are 144 and 2047, and the average for all locomotives, 544. The average data for the eight large mines in Tables 4 and 7 are respectively 111 and 1985 with a general average for all gathering and main locomotives of 497. These statistics would indicate that the duties imposed upon a locomotive in a single-haulage-system mine are heavier than those imposed on a locomotive in a double-haulage-system mine. This may be explained by the fact that the weight and capacity of all locomotives in the two-stage-haulage mines will average less than in mines having single-stage haulage.

Mine D

The daily production is 5000 tons. The mine cars weigh 2400 pounds empty and hold 3.50 tons of coal. The mine is developed uniformly, two-fifths of the production coming from the eastern section and three-fifths from the western. Two main-line locomotives operate in each section, each locomotive serving from three to five partings and averaging 26 trips of 15 cars each per shift of eight hours. The average length of main haul is nearly one mile. The system of train despatching minimizes the time lost on the partings, as the haulage-boss at the shaft bottom keeps in telephonic communication with a man stationed at each parting and, whenever a trip is reported as made up there, sends a locomotive to that point.

Tables 10 and 11 give the data obtained by a detailed study of Mine D, and Fig. 26 is a skeleton diagram of the haulage system at the time the observations were made. All main-line haulage is done by four trolley locomotives.

Table 10 gives the territories covered in gathering by fifteen locomotives and five mules. Two of the partings are served by both locomotives and mules. One point brought out by this tabulation is that although mules do not handle as great tonnage per day as the locomotives, they travel considerably farther. The mules in this mine average 8.5 miles of travel daily whereas the locomotives average but 4.4 miles.

Mine B

One-third of the production comes from the north side and two-thirds from the south side of the mine. Four 15-ton main-line locomotives haul an average trip of 15 to 16 cars over an average distance of 4630 feet, making slightly more than two round trips per hour. One 15-ton relay locomotive operates between two main partings and forms part of the main haulage system. Two other locomotives not only haul to the shaft bottom an average of six trips of eight cars each per day, but also gather the cars from the working faces. The grade on the main haulage road is generally in favor of the loaded cars and in some instances the grade is so steep that the loaded trips must be limited in size so that they can be safely handled by the locomotives.

The empty cars are taken to the working faces and the loaded cars obtained there by the gathering locomotives. In some of the rooms 4.5 per cent grades are encountered, thus taxing the gathering locomotives. The average capacity of the gathering partings is thirty cars, while the average main-line trip is 17 cars. Hence a supply of empties can be left on the gathering partings between main-line trips and the gathering locomotives need not wait for the return of the main-line trip before returning to the room faces.

Mine C

The mine is divided into four separate, nearly equal sections served by three main-line trolley locomotives, one hauling from each of two sections and one handling the tonnage of the other two sections.

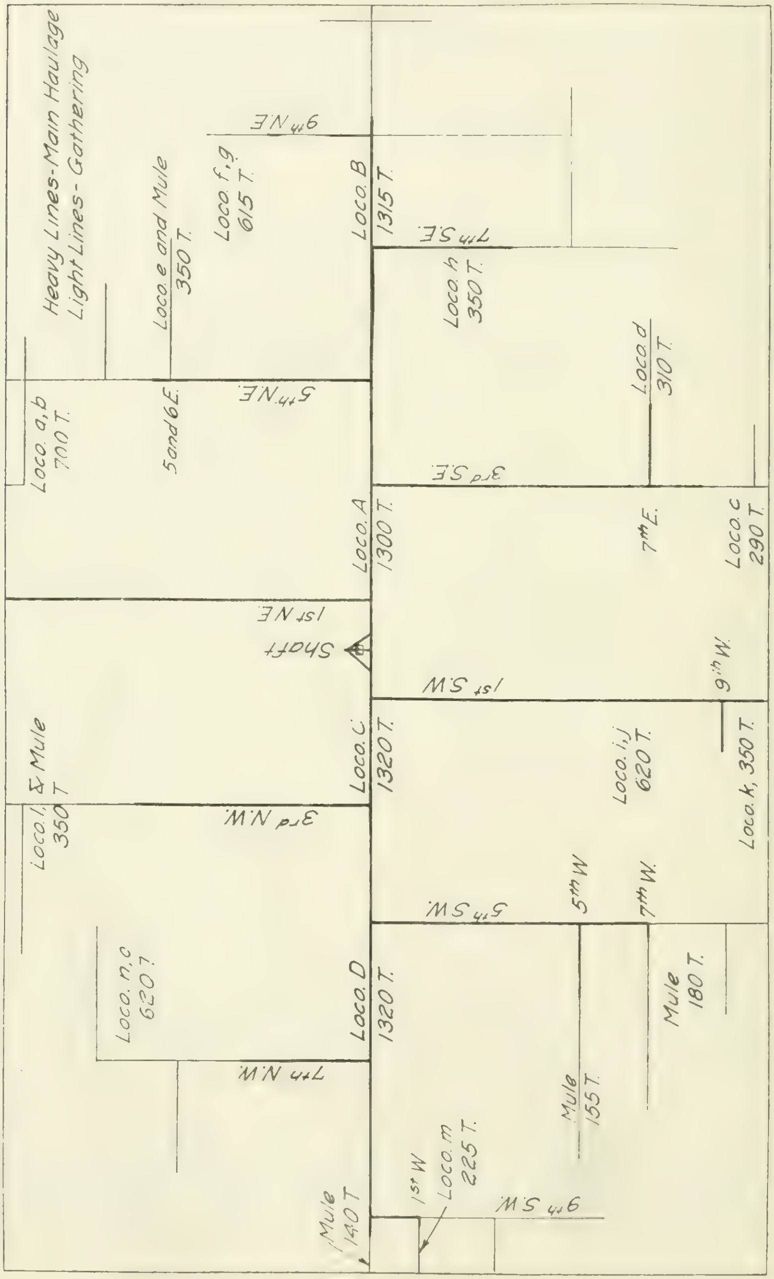

FIG. 26. HAULAGE DIAGRAM MINE D

TABLE 10. GATHERING HAULAGE—MINE D

Partings	Mules	Locomotives			Approx. Dist. Hauled Feet	Cars per Trip	Trips per Day	Cars per Day	Tons per Day	Ton-Miles Coal	Total Ton-Miles	Loco. Mi. or Mule Mi. per Day	No. Men in	
		No.	Wt.	Kind									Entries	Rooms
1st N. E.		(a) 1	6-ton	Reel	1500	12	8	96	700	95.4	160.8	4.55	0	24
		(b) 1	8-ton	St. Bat.	1250	13	8	104		86.2	145.3	3.79	10	16
3d S. E.		(c) 1	6-ton	St. Bat.	1800	11	8	80	290	98.9	166.7	5.45	0	22
7 E, 3d S. E.		(d) 1	6-ton	Reel	1400	12	8	90	310	82.2	138.6	4.24	0	24
5 & 6, 5th N. E.	1	(e) 1	8-ton	St. Bat.	1400	8	8	60	350	55.7	93.9	4.24	6	10
		....	...	...	800	2	20	40		21.2	35.7	6.05	4	6
9th N. E.		(f) 1	6-ton	St. Bat.	1000	14	8	105	615	69.6	117.3	3.03	6	22
		(g) 1	6-ton	Reel	1700	9	8	70		78.9	133.0	5.15	10	8
7th S. E.		(h) 1	8-ton	St. Bat.	1200	13	8	100	350	79.5	134.0	3.64	0	26
1st S. W.		(i) 1	6-ton	Reel	1500	11	8	85	620	84.5	142.5	4.55	4	18
		(j) 1	6-ton	Ree	1500	12	8	95		94.5	159.3	4.55	6	18
9W, 1st S. W.		(k) 1	6-ton	St. Bat.	2100	13	8	100	350	139.2	234.7	6.36	8	18
3d N. W.		(l) 1	6-ton	Reel	1500	13	8	100	350	99.4	167.6	4.55	10	16
	1	....	....	....										
7W, 5th S. W.	1	...	....	....	800	2	25	50	180	26.5	44.7	7.58	4	10
5W, 5th S. W.	1	....	....	....	1200	2	24	45	155	35.8	60.4	10.90	2	10
1W, 9th S. W.		(m) 1	8-ton	St. Bat.	1200	8	8	60	225	47.7	80.4	3.64	8	8
Main W	1	....	....	....	1200	2	20	40	140	31.8	53.6	9.08	6	4
7th N. W.		(n) 1	8-ton	St. Bat.	1800	10	8	80	620	95.4	160.8	5.45	8	12
		(o) 1	8-ton	St. Bat.	1100	13	8	100		72.9	122.9	3.34	4	22

Letters in parentheses refer to territories indicated on the diagram of this mine, Fig. 26.

Table 11

Main-Line Haulage—Mine D

Partings	Locomotives No.	Locomotives Wt.	Locomotives Kind	Approx. Dist. Haul'd Feet	Cars per Trip	Trips per Day	Cars per Day	Tons per Day	Ton-Miles Coal	Total Ton-Miles	Mileage per Loco.
1st N. E.	(A) 1	15-ton	Trolley	4400	14	14	200	700	1157	1943	51.6
3d S. E.				5700	10	8	80	290			
7 E, 3d S. E.				5300	14	7	90	310			
5th N. E.	(B) 1	15-ton	Trolley	4800	14	8	100	350	1186	1996	48.7
9th N. E.				5200	15	12	175	615			
7th S. E.				5400	15	7	100	350			
1st S. W.	(C) 1	12-ton	Trolley	4500	23	8	180	620	1081	1837	37.7
9W, 1st S. W.				4500	16	7	100	350			
3d N. W.				4000	14	8	100	350			
7 W, 5th S. W.	(D) 1	15-ton	Trolley	5700	15	4	50	180	1381	2411	56.7
5W, 5th S. W.				5500	16	3	45	155			
1 W, 9th S. W.				5900	16	4	60	225			
Main W.				6900	14	3	40	140			
7th N. W.				5500	15	12	180	620			

Letters in parentheses refer to territories indicated on the diagram of this mine, Fig. 26.

Grades are uniformly level. Each of the four sections of the mine has a separate current of air and a minimum number of self-closing, double doors. No trappers are employed.

Mine E

The two main sections of the mine furnish nearly equal production and two main-line locomotives operate in each section. In the northeast section the round trip averages over 3 miles.

Mine F

The east and west sections are laid out symmetrically and have about equal productions. One main-line locomotive operates in each section.

Mine G

The mine has two sections, east and west, which produce about equal amounts. There is one main-line locomotive for each section.

Mine H

There are two partings in the east workings and three partings in the west workings, each set of partings being served by one main-line locomotive.

Mine I

The eight partings in this mine are so grouped into pairs that all main haulage is performed by four trolley locomotives. An average of 500 tons of coal goes to the shaft bottom daily from each parting. Gathering in this mine is mixed, but not in the sense that both locomotives and mules are used in the same territories. Instead, 3 mules are used exclusively in gathering to one parting, while 14 locomotives handle the remaining seven-eighths of the mine. The mules haul but one car per trip.

26. *Mine Cars.*—In Illinois as in other coal-mining states many kinds of cars are in use even in mines of a single district, worked in the same seam, and with all conditions essentially common. When a particular type of pit-car is once adopted it is a difficult and expensive process to modify that type. It may thus happen that two neighboring mines, perhaps of common ownership, may be equipped with unlike cars and that in consequence the haulage and hoisting arrangements are so dissimilar that any interchange of cars for convenience or emergency is impossible. The regularity of the pitch, the thickness and the depth of the seam, the nature of the roof, the type of haulage system to be used, and the extent of the mining property all have weight in determining the design of mine cars.

Car Body

Formerly all coal-mine cars were constructed with wooden bodies, steel and iron being used in the wheels and axle and for stiffening the body. Indeed there are still some operators who, with strong arguments therefor, retain those wooden cars—and there are some mines which have both wooden and steel cars in service. General practice, however, is restricting usage to either type of car exclusively in any one mine, and the all-steel car is coming more generally into use and has now wholly superseded the wooden car in many mines. There are several reasons for this change.

One desideratum in the design and construction of a serviceable car is the utmost stiffness and strength in the trucks or running gear. The axles should always be in true alignment, and, since these members are held in their relative positions by their attachment to the floor of the car, stiffness in the car floor is of great importance. Without this rigidity there is a tendency for the car-wheels to climb the rails, frequently with derailment. The greatest stiffness is afforded by steel floors, and the all-steel car is much stiffer in this respect than is the car with a steel bottom but with wooden sides and ends.

Wreckage of mine cars occurs often at derailments. Experience proves that the all-steel car is the more resistant to injury or deformation, hence is less likely to be injured in such accidents and causes less delay to haulage. On the other hand, repair work on the steel car is frequently the more difficult and expensive.

There is no fixed ratio between the relative weights of the two general types of cars. Much depends upon the design, which in turn depends upon the methods used in dumping the loaded cars. A car used with rotary dumpers may be lighter in weight than one of equal capacity used with automatic-dumping cages. A steel car will be slightly less in width than an equivalent wooden car, this affording more clearance along the sides. The superior rigidity of the steel car causes it to travel more smoothly. Consequently there is less spillage of coal along the roads and loads may therefore be topped higher than is practicable with wooden cars. A steel car ordinarily has a longer life than a wooden car, but this feature is considerably offset by its greater initial cost. The expense of construction and upkeep may be less for steel cars than for wooden cars when the figures are distributed to cost per ton of coal ultimately handled per car.

Truck

A car truck comprises two axles with their bearings. Strength and minimum weight are prime factors in the design of the ideal truck, but frequently an axle is too small to withstand its imposed duty.

Recently the outside-journal bearing, similar to that used on railway cars, has been successfully used on heavy steel coal cars for slope or drift mines. These journals possess merit for such service; yet the spragging of such cars is difficult and hand brakes become necessary, thus rendering the cars poorly adapted to service in self-dumping cages.

Wheel-Base

The length of wheel-base in Illinois is ordinarily between 16 and 30 inches, the maximum being 42 inches. A 24-inch wheel-base is general for a car 7 to 8 feet long, and a 30-inch wheel-base for a car 10 feet long; while, when short turn-outs are necessary, a shorter wheel-base is used. An advantage of the short wheel-base is the greater ease with which a car may be re-railed, owing to a more easily balanced load. The increase of a couple of inches between axles may add many pounds to the weight lifted by the miner in re-railing a car. A long wheel-base is generally conducive to easy running and minimum derailments from cars climbing the rail.

Wheels

The car wheel should have as great a diameter as possible considering the distance between roof and rail and the capacity of the car. The greater the diameter of the wheel the less is the power required to move the car, but the net load carried may be less. The diameters used in Illinois vary from 14 to 20 inches. The tread of the wheel, to provide the best service, should be of chilled steel while the angle between the flange and the tread is approximately 100 degrees. Both plain and roller bearings are used extensively. Wheels are now often self-oiling regardless of the type of bearing, and the cost of lubrication is a feature that cannot be lightly ignored.

Bumpers and Couplings

In designing bumpers two factors control: the safety of the coupler, and the greatest mechanical efficiency. Thick, round, single bumpers with a single link-and-pin coupling seem to be most generally used in this state. Their advantages are that cars do not become locked on curves and that less slack is found between cars when a trip is starting or stopping. The twin-bumper car is also used in Illinois. The type of coupling used with the twin bumper varies. In some mines the gravity coupling has proved satisfactory under unusually severe conditions. Recent designs feature a spring drawbar with a link coupling that decreases the jerking of trips and the accompanying loss of coal. With this spring drawbar fewer cars are derailed at starting, trips require less power in starting, and the general wear and tear on the car is less. Partially offsetting these advantages are a greater initial cost and additional repair expenses.

Capacity of Mine Cars

The weight and capacity of a mine car are important items in connection with the handling of the car by men. If the cars are pushed to the face by the miners, the number of cars taken to the parting by the gathering locomotives is increased in proportion to the time saved by the locomotive in not having to run into the rooms. This time may amount to several hours per day and can be utilized by the locomotive in haulage on the entry.

If cars are taken to the faces by the locomotive they should have the maximum capacity for the given conditions as the time consumed in taking a single car to the face is no greater for a large than for a small car, and the larger the car the longer it remains in the room during loading. The largest car now used in Illinois contains about 5¼ tons.

Number of Cars Required

The number of cars in use in any mine is equal to the number of loaders plus a variable reserve. Although each man usually has a separate working place, two loaders generally work together in a room while the adjoining room is being undercut by a machine. Therefore, the car supply at the mine should include for each room where cars are being loaded one car in the process of being loaded and one in transit between the room and shaft bottom or drift mouth. In addition there should be a certain number of surplus cars on the shaft bottom or tipple landing and on the various partings, to prevent delays and to replace those undergoing repair.

The minimum empty-car reserve on the shaft bottom for each main-line section of the mine should be not less than the average number of cars per trip for that section. For instance, if there are two main-line locomotives hauling to the shaft bottom, one from each side of the mine, delivering to two tracks on the shaft bottom, and if the average trip for each side of the mine is 15 cars, there should be storage space for at least 30 empty cars on the shaft bottom, so that the incoming locomotive may find an empty trip ready to couple to as soon as it has uncoupled from its loaded trip. Likewise to prevent delay on each of the partings there should be a trip of loaded cars ready for the locomotive as soon at it has uncoupled from its empty trip. As shown in Table 3, the empty-car reserve varies widely at different mines.

The factors that influence the number of cars required for a given daily output are: the car capacity, the length of haul, the system of haulage, and the method of operating the haulage system. The greater the capacity per car the fewer the cars to be hoisted per day for a given output, and the longer the haul the more cars will be required. Mechanical haulage should require fewer cars than mule haulage as the cars are moving more rapidly and usually cars of larger capacity are used.

An unnecessary reserve of cars is objectionable on account of the extra capital they represent. The average life of a mine car is five to eight years and, assuming cost of the cars as $160, about $20 per car must be charged off each year for a life of 8 years. According to Table 3, a car is loaded on an average 2.05 times per day and the average capacity is 3.64 tons per day. In a year, or 200 days, a car will handle 1492 tons. On this basis the mine-car depreciation charge is approximately 1.34 cents per ton of coal hauled. This does not include the cost of repairs but only the gross depreciation per car of $20 per year.

For a simple calculation consider a mine having four main partings of equal production which for a given day will supply an equal number of cars, say 300 each, or a total of 1200 cars for a mine producing between 4000 and 5000 tons. If the "turn" runs four cars per loader, 75 loaders will furnish 300 cars from each of four districts or 300 loaders will be required for the total. Consider one unit of this group, and assume that 75 loaders will produce 300 cars and that the parting is 5000 feet from the shaft bottom. Assume also one main locomotive under ordinary operating conditions, 16 trips of 19 to 20 cars each at the rate of one trip every half hour, on the basis of ten minutes running time each way and ten minutes for switching at the two terminal points, this being an easy operating schedule. Under favorable conditions a gathering locomotive should deliver at the parting 100 cars per day, assuming the same number of trips per day as for the main-line locomotives, a running time of 6 minutes, a switching time of 24 minutes, and 6 to 7 cars per trip. This is, however, considerably higher than the average Illinois gathering performance. At most mines four locomotives are required to handle 300 cars as assumed, operating on the same schedule with five-car trips, and gathering approximately 75 cars per day per locomotive. On a practical operating basis with a fixed running schedule of one trip every half-hour, the empty reserve on the parting is zero, as the times of arrival of main and gathering locomotives should be within a few minutes of each other, just as in the schedule operations of ordinary town or city trolley cars. The total empty reserve in such instances is held practically on the shaft bottom.

There should be some possible combination of the "turn" and the number of rooms per panel that will make it possible to determine the frequency of trips and the number of cars per trip for the most economical gathering schedule. Cars should be distributed at uniform intervals throughout the day. If the "turn" is to be 8 cars for two men then a trip every hour should be regularly established. The larger the car, the longer the time required for loading and the fewer the cars required for the "turn." If the car is of such capacity that a six-car "turn" gives the desired tonnage, a trip every 80 minutes will be adequate.

Standardization

With the object of bringing about more uniformity in design and construction, efforts have been made to establish acceptable standard specifications for a few of the main features of coal-mine cars. A committee of the American Mining Congress for The Standardization of Underground Transportation Equipment, coöperating with The Industrial Car Manufacturers' Institute, has recommended the following specifications for the design and construction of coal-mine cars:

(1) A track gauge of 42 inches should be adopted for all new coal-mining developments.

(2) The most desirable wheel-base is 42 inches.

(3) The overall length of a car-body should be three times the wheel-base, thus making the standard length 126 inches, or 10 feet 6 inches.

(4) Standardized automatic couplings, comparable to those of surface railways, should be used. For a car with 16-inch wheels the center of such a coupling should be 10 inches above the top of

rail, with a variation of 1 inch above to accommodate 18-inch wheels and of 1 inch below for 14-inch wheels.

Discussion of these features elicited the following statements: About 80 per cent of all new track-work in coal mines of this country is of 42-inch gauge. This gauge will fit all mine conditions and will accommodate any appropriate car-body. A 42-inch wheel-base is theoretically correct and practical; it minimizes derailment, increases speed possibilities, and tends to lengthen the life of cars.

Repairs

The expense of maintaining mine cars is not generally known; hence the following data upon this matter gathered by one large Illinois coal-mining company are of interest. During a period of eight months or 117 operating days there were 400 cars in service. The average weight of an empty car was 2000 and the average load of coal per car, 5000 pounds. The total tonnage hauled was 292 877, with the daily average per eight-hour shift, 2503. There were used 9279 board feet of oak, besides bolts and washers. At this mine one carpenter would finish all the repairs to a two-ton wooden car in from 8 to 16 hours. The average life of a car was 5 years.

27. *Track Construction.*—Proper track construction and maintenance are important in any haulage system, as the expected benefit from expensive rolling equipment may be offset by a poor track. In many mines the defects in track construction would be much more apparent if the track could be lifted out intact and reproduced in all its variations on the surface. In development work track of a temporary nature only is laid. In the rooms where track is intended only for locomotives with one or two cars moving at a slow speed light construction is used. But on the main-haulage track the construction should be designed for the heavier locomotives and longer trips of cars that are now generally used.

Gauge

The gauge of track has a direct bearing upon the capacity of coal-mine cars. A narrow gauge permits a longer wheel-base on sharp curves but as a rule the car is subject to more derailment. Gauges varying from 36 to 42 inches are common in Illinois bituminous mines. The maximum gauge in the state is 48 inches. Other conditions being equal, the wider the track gauge the wider may be the car and the greater its capacity. In low coal this is a pertinent factor. Good roof conditions permit wide gauges. Entries are usually driven 12 feet wide and the 42-inch gauge has proved well adapted to such entries.

Rails

Until a few years ago 40-pound rails on the main entries, 30-pound on the cross-entries, and 20-pound in rooms were considered adequate; but with the advent of larger cars, heavier locomotives and longer trips, the weight of rails and sizes of ties and spikes have increased considerably. Numerous large companies have adopted as standard not less than 50-pound rails for the main entries and 30-pound for rooms and cross-entries. In some instances 60- to 70-pound rails are used on main entries with excellent results. Where large cars and gathering locomotives are used 30-pound rails possess considerable advantage over 20-pound rails for rooms and cross-entries, as the repair cost for the heavier rail is but slightly more than that for the lighter, while the added initial expense—both for the material and labor—is usually justified by the rails lasting longer.

Ties

Timber ties of the following sizes are generally used: main entry, 5 x 6 inches; cross entry, 4 x 5 inches; rooms, 3 x 4 inches or 4 x 5 inches. Oak is used if obtainable, although considerable quantities of elm, hickory, and sassafras are consumed. Hewn ties with the bark removed are generally used, and are spaced 18 inches in entries and 36 inches in rooms. The sizes of spikes used are: for 40 or 50-pound rail, ½ x 4 inches; for 30-pound rail, ½ x 3½ inches; for 20-pound rail, ½ x 2½ inches.

Steel ties are used to a limited extent both for entries and rooms. Their advantages are:

(1) They afford additional height of from 2 to 4 inches above the rails, thus permitting the use of higher coal cars and heavier loading of low cars.

(2) The ties are lighter in weight and more easy to handle than timber ties, therefore are more readily laid and taken up.

(3) The rails being held by lugs the track is easily kept at a true gauge and the spreading of rails is prevented.

(4) Because of the less height of rail from the ground a derailment can be more readily remedied.

Switches

Switches may be classed under three general heads according to the method of operation:

(1) Ground track, in which the lever lies close to the ground and moves either parallel to or at right angles to the track.

(2) Switch stand, in which the lever moves either perpendicular to the track or rotates.

(3) Automatic or partly automatic, in which the lever is thrown by contact with a locomotive, or by motormen or trip riders without leaving the locomotives.

All switches for main haulage roads should be substantial and reliable. To avoid wrecks and to properly care for the rolling stock, the lead should be as long as possible and should be definitely calculated. Manufacturers of track equipment recommend a 4- to 6-foot switch-point to be used with a No. 4 frog, the length depending on the length of locomotive wheel-base. Some operators prefer the kick latch in place of the switch-point operated by the switch-stand. If the roads are kept clean the kick latch has some advantages on secondary haulage roads.

The cost of switches varies with the design. The following represents average requirement of material and labor and the costs will vary with their fluctuations. The materials will be: one No. 4 riveted frog with 6-foot switch-points; one switchstand with bridle and connecting rods complete; forty 5 x 6 inch ties; sixteen ½ x 4 inch spikes, and eight special bonds and wiring. The labor will include delivering material to place, cost of laying the switch and bonding.

If gathering motors are used, the room turnouts are similar to the main-line switches upon a smaller scale. Good practice at the larger mines demands the use of the maximum radius, which varies usually from 25 to 35 feet. These radii are very nearly those used with No. 2 and No. 2½ frogs. If it is desirable to carry the track along the rib it may be necessary to change the radius of the curve entering the room. It can readily be seen that it would be impossible to enter a room neck 10 feet wide with a turnout of 25-foot radius if the neck were driven at right angles to the entry.

A room switch costs much less than a turnout or parting switch because of the smaller size of the rail and its shorter length. The salvage value is greater because it is more quickly removed. Cast-steel frogs are used at present for secondary haulage and have proved entirely satisfactory; their chief advantage being their low initial cost, and their chief disadvantage the difficulty of holding the frog in place.

With mule haulage the switch is much simpler and consists of one frog-point, one straight rail, and one turn rail. The length of lead will vary between 8 feet 6 inches and 12 feet 6 inches, depending on the gauge and the wheel-base of the car. Empty cars enter rooms by being slewed over the open fixed point by the driver. On the return the loaded car is shunted to the entry road with little jar.

V. Underground Haulage Costs

28. *Cost Accounting.*—Copies of the cost-accounting sheets of sixteen well known companies were studied for the purpose of constructing a table that would show how the companies itemize their haulage costs but, owing to the lack of any uniformity in this practice, the tabulation proved impossible. Some companies maintain no special account for Haulage but place all wages for this branch of mining under General Expense. Companies frequently include haulage and hoisting under Transportation, with so few sub-items as to prevent analysis.

Cagers are charged by four companies to Haulage, by three to Hoisting, by five to General, by one to Transportation, and by one to Caging, while six companies do not carry this item. Four accounts place tracklayers under Haulage, six under General. Trappers are charged to Haulage by but one company, this occupation being usually charged to Ventilation or General. The only occupations that are uniformly charged to Haulage are switchmen, greasers and sand driers. Of the 13 companies that itemize trip riders, 11 consider them as chargeable to Haulage.

It is even more difficult to secure cost under the subdivisions of shaft-bottom haulage, main-line haulage, and gathering haulage—the three general divisions into which mine haulage may logically be divided and which are necessary for a satisfactory comparison of details. Hoisting and haulage are often combined. The hoisting cost is, however, small in comparison with the haulage cost and it is more nearly uniform for different mines than is the haulage cost.

29. *Standardizing Cost Accounts.*—The lack of uniformity in the accounting of coal-mining costs applies not only to haulage but to every other phase of the industry, as operating companies naturally object to the radical changes in bookkeeping necessary for the adoption of a universal system. Thus, close comparison of expenses and profits of companies operating under either similar or dissimilar natural or commercial conditions—a study that would yield informa-

tion of value in dealing with commercial and industrial problems—has been impossible.

The Committee on Standard System of Accounting and Analysis of Cost Production of the National Coal Association has prepared the following schedule of the natural subdivisions of the work in and around a coal mine:

1. Mine Office
2. Superintendence
3. Engineering
4. Mining
5. Timbering
6. Deadwork
7. Tracklaying
8. Drainage
9. Ventilation
10. Haulage and Hoisting
11. Dumping and Tallying
12. Preparation
13. Railroad Car Loading and Yard Expense
14. Power
15. Repairs to Buildings and Permanent Structures

This same committee explains that haulage and hoisting should be accounted as follows:

Generation and Transmission of Power

This item includes the proportion of expense of generating power chargeable to haulage and the construction and upkeep of transmission lines and haulage circuits.

Care and Maintenance of Equipment

This item covers:

(a) Hoisting and haulage engine repair parts, lubricants, packing and waste, and wages of hoisting engineer and mechanics employed in care and repair; hoisting and haulage ropes, cage repairs, and replacements; safety devices, guides, and sheaves.

(b) Care and maintenance of motors; when motor haulage is used, repair parts and labor of care and repair.

(c) Care and maintenance of pit-cars; labor and material used in keeping pit-cars in repair; new cars to replace wrecked or worn-out cars, and additional cars necessary to maintain output by reason of increasing length of haul after mine has reached its contemplated output capacity.

(d) Care and maintenance of live stock, harness, stable supplies, grain and hay, wages of stablemen and veterinary, clipping and shoeing, etc.

Conducting Transportation

This item includes wages of drivers, boss drivers, motormen, trip riders, couplers, cagers, pushers, oilers, trappers, switch throwers, jackmen, and that part of hoisting engineer's wages not charged to Maintenance and Repairs.

Maintenance of Way

This item includes repairing roads, cleaning roads, relaying track, also new ties, rollers for rope haulage, etc.

Under the head of Tracklaying the committee report says:

> "While track is immediately connected with and necessary for the transportation of coal to the shaft bottom, and hence a necessary item incident to Haulage, it has long been regarded as a significant item in the cost sheet, and should stand by itself. To this account should be charged rails, ties, spikes, and fastenings, and the labor of grading roads and tracklaying in advancing work. Repairs to track should be charged to Haulage and Hoisting under Maintenance of Way. Purchases of track material should be charged to Track Material Account, and as the material is taken into the mine it should be credited and charged Tracklaying."

The committee's explanation of the item Tracklaying (usually called Trackwork by operators) illustrates a common reason for disagreements between haulage costs as estimated by various companies. As noted above, rails, ties, etc. for advancing roads are charged under a separate item Track, while relaid track is a part of haulage under Maintenance of Way. If track is pulled out of an entry and used in a new entry is it to be considered relaid and chargeable to Haulage or as advancing work and chargeable to Tracklaying? This is merely an instance of the difficulty of defining any system of segregated items so clearly that it is not open to misinterpretation.

In its wartime collection of costs the Federal Trade Commission asked for haulage costs under the following heads:

Haulage:
  Animal
  Mechanical
  Equipment Repairs
  Stable Expense
  Labor
  Supplies
  Total

The following items in the Instructions for Compiling Coal-Mining Costs have direct bearing upon haulage:

> *Labor—Haulage.*—This account shall include the wages of hoisting engineers, cagers (top and bottom), motormen, brakemen, trip riders, switchmen, couplers, greasers, spraggers, stable boss, drivers, sand dryers, and other labor employed to operate the haulage facilities other than standard gauge railroad equipment. Wages of employees, such as electricians, blacksmiths, trackmen, car and locomotive repair men, and men engaged in maintaining haulage equipment and tracks, shall be charged to Maintenance Account.
>
> *Maintenance and Repair.*—This account shall include the cost of labor employed in repairing and maintaining (1) the tipple, powerhouse, tracks, and other mine structures; (2) mining machines, pumps, fans, boilers, engines, motors, locomotives, mine cars, and other mining equipment.
>
> *Feed and Other Stable Supplies.*—This account shall include the cost to the operator of feed, bedding, and other stable supplies.
>
> *Supplies—Maintenance and Repairs.*—This account shall include the cost (1) of supplies used in maintaining and repairing the tipple structure, powerhouse, and other mine buildings and structures, and (2) of supplies and parts used in repairing mining machines, pumps, fans, boilers, engines, motors, locomotives, mine cars, tracks, and other mining equipment.

Here are four separate items for haulage, any one of which might be quoted from a government publication and be misleading as covering only part of the haulage costs.

30. *Itemized Haulage Costs for Typical Large Illinois Mines.*—Table 12 gives transportation costs for twelve mines itemized as suggested by the Coal Association except that the cost of hoisting has been deducted when possible. Two of these mines G and H are also listed in Table 3 of shaft-bottom costs. In general the average daily production of the mines in Table 12 is less than of those in Table 3. The costs per ton in Table 3 apply only to the shaft-bottom labor, whereas the costs in Table 12 cover all haulage labor.

TABLE 12

HAULAGE COSTS AT TWELVE REPRESENTATIVE ILLINOIS COAL MINES

Items	1		2		3		4	
	Labor	Material	Labor	Material	Labor	Material	Labor	Material
(a) Conducting Transportation	$35 737.12	$ 993 20	$ 7 778.49	....	$23 786.80	$ 5 355.33	$16 797.91	$1 522.00
(b) Maintenance of Way	12 238.86	5 997.62	1 028.83 6 566.22	$2 546.34	13 328.05	13 468.21	9 354.47	6 286.98
(c) Generation and Transmission of Power	2 826.92	5 756.42	766.20	....	4 898.54	not given	3 010.51	not given
(d) Care and Maintenance of Equipment	7 302.68	.....	1 149.45	681.36	3 048.45	not given	1 636.88	not given
Totals	$58 105.58	$12 747.24	$17 289.19	$3 227.70	$45 061.84	$18 823.54	$30 799.77	$7 808.98
Total Cost	$70 852.82		$20 516.89		$63 885.38		$38 608.75	
Total Cost per Ton	34.10c		25.70c		30.44c		30.65c	
Labor Cost per Ton	28.00c		21.70c		21.47c		24.45c	
Total Tonnage	207 710		79 817		209 889		125 971	
Period	6 mo. 1920		1 mo. 1920		3 mo. 1920		Aug.-Oct. 1920	
Average Daily Tonnage	2600		3400		3100		1700	
System of Haulage:								
Main	2 Trolley Loco.		2 Trolley Loco.		4 Trolley Loco.		3 Trolley Loco.	
Gathering	14 Reel Loco.		14 S. B. Loco.		12 S. B. Loco., 7 Mules		6 S. B. Loco., 7 Mules	
Weight Tons Coal per Car	3.30		3.25		3.15		3.00	

TABLE 12 (CONTINUED)

HAULAGE COSTS AT TWELVE REPRESENTATIVE ILLINOIS COAL MINES

Items	5		6		7		8	
	Labor	Material	Labor	Material	Labor	Material	Labor	Materia
(a) Conducting Transportation	$ 9 426.32	$ 650.06	$285.58	....	$ 7 602.28	$ 875.06	$ 6 936.24	....
(b) Maintenance of Way	10 781.47	2 867.57	139.20	3c ton	1 277.79 9 428.36	1 152.55	2 912.98 3 758.59	$1 308.78
(c) Generation and Transmission of Power	180.75	204.83	63.08	7c ton	238.34	not given	248.59	....
(d) Care and Maintenance of Equipment	724.11	820.07	183.03	3c ton	3 508.53	not given	350.90	....
Totals	$21 112.65	$4 542.53	$670.89	13c ton	$22 055.30		$14 207.30	....
Total Cost	$25 655.18		$804.89		$25 924.54		$16 120.98	
Total Cost per Ton	31.62c		44.70c		32.40c		37.40c	
Labor Cost per Ton	26.02c		31.70c		27.60c		33.20c	
Total Tonnage	81 134				79 824		43 013	
Period	Nov. 1920		Aver. day 1920		30 days 1920		Oct. 1920	
Average Daily Tonnage	4000		1800		3200		2250	
System of Haulage:								
Main	4 Trolley Loco.		3 Trolley Loco.		3 Trolley Loco.		3 Trolley Loco.	
Gathering	16 S. B. Loco., 4 Mules		7 S. B. Loco.		15 Reel Loco., 9 Mules		22 Mules	
Weight Tons Coal per Car	3.00		2.70		3.25		2.25	

TABLE 12 (CONCLUDED)

HAULAGE COSTS AT TWELVE REPRESENTATIVE ILLINOIS COAL MINES

Items	9		10		11		12	
	Labor	Material	Labor	Material	Labor	Material	Labor	Material
(a) Conducting Transportation	$ 5 173.81	......	$ 9 453.36	$ 927.17	$11 967.20	$1 469.40	$ 9 347.52	$1 806.07
(b) Maintenance of Way	2 503.76 2 224.31	$1 431.71	4 692.88	1 065.57	3 756.11	3 451.86	2 550.35	2 752.12
(c) Generation and Transmission of Power	252.39	....	2 186.54	2 614.95	1 416.88	2 667.06	817.33	3 463 94
(d) Care and Maintenance of Equipment	1 512.19	2 118.68	3 204.43	2 018.26	1 471.25	1 009.30	1 894.72	1 728.96
Totals	$11 666.46	$3 550.39	$19 537.21	$6 625.95	$18 611.44	$8 597.62	$14 609.92	$4 751.09
Total Cost	$15 216.85		$25 163.16		$27 209.06		$24 361.01	
Total Cost per Ton	58.70c		62.70c		77.80c		74.80c	
Labor Cost per Ton	45.00c		48.70c		53.20c		41.90c	
Total Tonnage	25 907		40 115		34 972		32 544	
Period	Oct. 1920		Oct. 1920		Oct. 1920		Oct. 1920	
Average Daily Tonnage	1780		1500		2200		1770	
System of Haulage:								
Main	2 Trolley Loco.		4 Trolley Loco.		4 Trolley Loco.		4 Trolley Loco.	
Gathering	8 Reel Loco.		49 Mules		43 Mules		36 Mules	
Weight Tons Coal per Car	3 25		2.00		2.25		2.20	

The effect of daily production on shaft-bottom costs is shown by Table 3, for the average labor cost per ton in the six largest mines is 26.33 cents while in the six smaller mines it is 41.32 cents. Similarly the average is 39.51 cents as against 58.22 cents in the six smaller mines.

As shown by Table 3, there is cost advantage in handling large cars on the shaft bottom; not only is this cost lowest where the largest cars are used and highest where the smallest cars are used—which might be accidental—but the average cost in four mines where cars holding 4 tons and over are used is 1.31 cents per ton, whereas in 6 mines where cars holding less than 4 tons are used it is 1.51 cents per ton.

In Table 13 hoisting costs are given for comparison with haulage costs. The effect of tonnage upon costs is more pronounced in hoisting than in haulage.

TABLE 13

HAULAGE COSTS AT FIVE MINES OF COMMON OWNERSHIP

In Cents per Ton

Items	Period	I	II	III	IV	V	Averages
Gathering . . .	December, 1920	10.73	10.55	10.61	9.88	13.63	11.08
Main haulage .		4.16	2.67	5.80	3.89	4.07	4.12
Track work . .		4.42	13.91	8.08	17.06	8.03	10.30
Total haulage		19.31	27.13	24.49	30.83	25.73	25.50
Tons mined . .		47 602	88 018	41 702	73 758	37 928	57 802
Hoisting		1.47	1.09	2.04	1.30	3.81	1.94
Days worked		25.00	26.00	26.00	26.00	25.00	25.60
Gathering	May, 1921	10.60	11.57	12.47	8.73	26.12	13.90
Main haulage .		4.05	2.20	6.78	3.20	5.59	4.36
Trackwork		4.98	13.54	9.98	8.88	4.65	8.41
Total haulage .		19.63	27.31	29.23	20.81	36.36	26.67
Tons mined		28 921	55 951	28 541	61 121	4 418	35 790
Hoisting		2.44	1.72	2.58	1.89	15.87	4.90
Days worked . .		13.00	15.00	18.00	20.00	3.00	14.00

In Mines I and V gathering is by mules; in the other mines it is by storage-battery locomotives. All main haulage is by trolley locomotives.

The sub-items considered in obtaining the above costs are as follows:

Trackwork—Tracklayers, helpers, handling track material, grading track (laborers).

Gathering—Mule feeder, blacksmith (part time), drivers, veterinary service, boss driver, motormen on gathering motors, trip riders on gathering motors, battery charger, electrician (part time), labor on repairs, trappers, and flaggers.

Main—Motormen, trip riders, electrician (part time), switch thrower, wiremen (part time), other labor on repairs, trappers and flaggers.

The following are labor costs chargeable to Haulage, exclusive of generation and transmission of power, at an Illinois mine which produces an average of 5000 tons daily:

1 cager	$ 7.50	
2 cager helpers	14.50	
2 blockers	14.50	
2 couplers	14.50	
2 oilers	14.50	
1 switcher	7.25	
2 sump cleaners	14.50	
7 drivers	52.50	
19 motormen	152.76	
19 trip riders	142.50	
19 tracklayers	142.50	
16 track helpers	116.00	
11 repairmen	80.00	
4 cleaning falls	30.00	
7 brushing	50.75	
1 sprinkling roads	7.50	
1.5 cleaning roads	12.50	
3 hauling dirt	22.50	
1 mule feeder	7.50	
2 electric bonders	15.00	
4 locomotive repairmen	30.00	
Total	$949.26	
Haulage wage per ton	18.99	cents

Labor, delivery of material and supplies:

1 cager	$ 7.50	
1 cager helper	7.25	
5 drivers	37.50	
Total	$52.25	
Daily wage per ton	1.04	cents
Total haulage labor cost per ton	20.03	cents

At another large mine with an average daily output of 5200 tons the average number of employees engaged in haulage operating and in maintenance of way, and their total wages, are as follows:

3 haulage bosses	$ 24.18
29 motormen	233.74
29 trip riders	217.50
32 trappers	128.00
6 jackmen	43.50
6 repairmen	45.00
2 electricians	15.00
1 oiler	7.25
33 tracklayers	247.50
27 track helpers	195.75
18 timbermen	130.50
8 road cleaners	58.00
1 sprinkler	7.50
2 bonders	15.00
4 cagers	29.50
6 blockers	43.50
1 switcher	7.25
2 couplers	14.50
Total	$1463.17
Total haulage wage cost per ton	28.14 cents

At one mine producing 4500 tons daily, the following items cover the daily labor costs of maintaining and conducting haulage:

Maintenance:

1 chief electrician, half time	$ 6.50
1 shop foreman, half time	5.50
1 motor charger, half time	7.50
1 sub-station attendant, half time	4.52
1 motor oiler	7.50
1 electrician	7.50
23 tracklayers at $7.50	172.50
16 track helpers at $7.25	116.00
6 road cleaners at $7.25	43.50
2 sump cleaners at $7.25	14.50
4 car repairers at $7.50	30.00
1 oiler	7.50
Total	$423.02
Total maintenance wages per ton	9.4 cents

Conducting:

16 motormen at $8.06	$128.96
16 trip riders at $7.50	120.00
1 motor boss	11.00
1 cager	7.50
2 spraggers at $7.25	14.50
1 coupler	7.25
1 switcher	7.25
Total	$296.46
Total conducting wages per ton	6.6 cents

This represents high efficiency. No trappers are employed. Double swinging doors are opened automatically by the locomotives, but the item of maintenance of doors is not included in cost.

At this same mine the electric-power costs for haulage for the year 1919 were:

	Kw-hr.
Total kw-hr. used by trolley locomotives	130 210
Total kw-hr. charging battery locomotives	129 650
Total kw-hr. for haulage	259 860

The cost of this power at the rate of 3.5 cents per kilowatt-hour was $9094.10. During the year the mine produced 373 847 tons of coal, thus making the power cost for haulage about 2.4 cents per ton. The complete haulage costs, not including any materials, were per ton:

	Cents
Wages for maintenance	9.4
Wages for conducting	6.6
Power	2.4
Total	18.4

Table 14 details the haulage costs in cents per ton for two large Illinois mines during January and February, 1921.

These two mines operate under similar natural conditions. Mine A has the longer hauls and there is no division between main and gathering haulage, whereas Mine F has two-stage haulage. Although general conclusions must not be drawn from these two mines nor for such a short period, the marked difference in cost between the two systems suggests the desirability of a more extended study of the two systems.

Table 14
Haulage Costs at Two Illinois Coal Mines
In Cents per Ton

Item	Mine A		Mine F	
	Jan.	Feb.	Jan.	Feb.
Tonnage for month	92 560	62 188	88 072	45 383
Tonnage Daily	4 500	4 480	3 500	3 700
Occupations:				
Motor Bosses	0.63	0.93	0.32	0.62
Motormen	3.54	3.36	4.64	4.43
Trip Riders	2.63	2.64	4.02	3.83
Couplers	0.26	0.29	0.40	0.38
Cagers	0.42	0.48	0.83	0.86
Other Bottom Men	1.16	1.12	0.79	0.72
Flagmen	1.71	1.59	4.26	3.35
Jackmen	0.17	0.17	....	0.13
Maintenance of Way	3.35	2.32	2.68	2.01
Trappers	1.06	0.89	0.50	0.67
Track Bosses	0.26	0.36	...	....
Tracklayers	2.70	2.02	7.21	7.03
Track Helpers	2.35	1.51	4.46	5.03
Totals	20.24	17.68	30.71	29.06
Repairs:				
Labor on Mine Cars	1.92	2.19	3.22	3.31
Supplies for Mine Cars	2.17	2.95	1.07	2.31
Totals	4.09	5.14	4.29	5.62
Labor on Locomotives	1.36	1.43	2.09	2.60
Supplies for Locomotives	1.14	2.38	2.79	2.58
Totals	2.50	3.81	4.88	5.18
Total Cost per Ton Exclusive of Power and Track Equipment	26.83	26.63	39.88	39.86

Table 15 gives the haulage employees and the total labor costs in cents per ton for four large Illinois mines.

The following estimates of haulage wages are for mines having electric haulage exclusively. The figures cited as ranges of costs per ton are distributed under four items. Main-haulage and general wage costs are about equal and each is about double the shaft-bottom wage cost per ton. The labor cost of gathering haulage usually equals or exceeds the sum of the three other items.

Shaft-Bottom Haulage . . . . 1 to 3 cents
Cagers, spraggers, blockers, couplers, car distributors, and all other employees handling cars on shaft bottom only

Main Haulage . . . . 2 to 6 cents
Motormen, trip riders, trappers, trackmen, timbermen, wiremen, road cleaners, switch throwers, etc., engaged directly on main haulage.

Gathering Haulage . . . . 8 to 15 cents
All employees engaged in hauling coal on the inside divisions, including motormen, trip riders, drivers, trappers, and all trackmen and such timbermen as are necessary for maintenance of way.

General . . . . 2 to 6 cents
All employees connected with haulage as a whole, as oilers, electricians (unless strictly main haulage), repairmen, sump cleaners, etc.

Total operating haulage labor cost thus may vary between 13 cents and 30 cents per ton. As the tonnage varies at a given mine on different days there will be a variation in the daily haulage cost per ton, even with the same working force.

Table 15

Haulage Labor Costs at Four Large Illinois Coal Mines

In Cents per Ton

Occupations	Mine 1		Mine 2		Mine 3		Mine 4	
	Main	Gather.	Main	Gather.	Main	Gather.	Main	Gather.
Motormen	3	12	4	2	3	15	4	14
Trip Riders	3	12	4	2	3	15	4	14
Tracklayers	11	16	2	8	1	21	7	15
Track Helpers	...	16	2	8	...	...	7	15
Switch Throwers	....	...	...	5	...	...	1	...
Wiremen	....	....	1	...	...	...	...	...
Trappers	....	....	4	9	....	...	...	14
Timbermen	..	....	4	2		.	...	.
Drivers	....	...	...	12	...	...	...	...
Batterymen	....	2	.	...	..	..	.	4
Mule Feeders	....	...	..	..	.			2
Road Cleaners	1	....	....	....	.	.	..	1
Totals	18	58	21	48	7	53	23	79
Total Cost Labor	$131.41	$437.48	$144.41	$283.08	$54.12	$405.10	$172.10	$542.27
Tonnage Daily	4500		3800		3400		4000	
Labor cost per ton:								
Main	3.0	....	3.8	.	1.6	.	4.3	..
Gather	....	9.7	...	7.4	....	12.0	....	13.5
Total per Ton, cents	12.7		11.2		13.6		17.8	

VI. Haulage Accidents

31. *Haulage Fatality Statistics.*—Table 16 gives the coal mine haulage fatalities in the United States and in Illinois for the period 1901 to 1920 inclusive, together with the average percentage of all fatalities for each five-year period. For the past ten years haulage fatalities have been second in importance only to those from falls. These two classes, which make up from 60 to 70 per cent of the number of deaths underground, occur for the most part singly or in small groups, hence do not attract public attention to the same extent as do explosions, which are third in importance. The number of deaths from falls is remarkably uniform year after year, forming almost 50 per cent of the total fatalities. Haulage deaths have been constantly increasing in per cent of the total and therefore should be given more attention as they seem to a great extent to be preventable.

In Illinois the percentage of deaths from falls of roof and pillar coal approximates that for the United States but the percentage of deaths from haulage is higher and shows a decided increase during the past decade. Such haulage fatalities are due not only to mine cars and locomotives but also to electricity and animals as shown in Table 17.

Table 16

Coal Mine Fatalities due to Haulage

By Five-Year Periods

Period	United States			Illinois		
	Total Underground Fatalities	Haulage Fatalities	Per Cent of Total	Total Underground Fatalities	Haulage Fatalities	Per Cent of Total
1901–1905	8 428	1097	13.0	668	84	12.6
1906–1910	12 017	1649	13.7	1024	145	14.2
1911–1915	11 424	1939	17.0	753	191	25.4
1916–1920	10 771	2201	20.4	904	278	30.8
Totals 1901–1920	42 640	6886	16.1	3349	698	20.8

Table 17

United States Coal-Mine Fatalities due to Haulage Classified as to Causes

Causes	Year				
	1916	1917	1918	1919	1920
1. Mine Cars and Locomotives:					
Switching and Spragging	17	6	15	12	6
Coupling Cars	12	7	11	6	13
Falling from Trips	43	29	36	17	26
Run over by Car or Locomotive	147	187	203	149	163
Caught between Car and Rib	87	122	113	105	98
Caught between Car and Roof	12	20	27	23	18
Runaway Car or Trip	42	67	68	42	43
Miscellaneous	30	50	33	27	38
Totals	390	488	506	381	405
2. Electricity:					
Direct Contact with Trolley Wire	66	46	55	39	29
Bar or Tool Striking Trolley Wire	5	2	4	2	3
Contact with Locomotive Parts	1	4	1	2	3
Totals	72	52	60	43	35
3. Animals	8	9	8	2	4
Total Fatalities Chargeable to Haulage	470	549	574	426	441
Total Fatalities Due to Coal Mining	2226	2696	2580	2317	2260
Per Cent Due to Haulage	21.1	20.4	20.2	18.4	19.7

Even falls are frequently caused initially by derailed cars knocking out roof supports.

Table 17 gives the classification of the causes of haulage fatalities in the United States for the five-year period, 1916-1920.

32. *Haulage Accidents in Illinois.*—Table 18 gives a more detailed causal analysis of haulage accidents for Illinois, and the accompanying graph, Fig. 27, shows the variation of the percentages of haulage to total fatalities throughout the period 1902-1921. The latter half of the period is fairly indicative of present operating conditions. For the past ten years haulage fatalities have averaged 27 per cent of the whole. During the years 1918 to 1921 inclusive the average number of employees in Illinois coal mines was 88 274 per year. These

Table 18
Causal Analysis of Haulage Fatalities in Illinois

Causes	Period					
	1902–05	1906–10	1911–15	1916–20	1921	Totals
Switching and Spragging	5	8	5	7	1	26
Coupling Cars	1	1	5	13	3	23
Falling from Trips	11	39	31	22	3	106
Run over by Car or Locomotive	12	15	29	72	24	152
Caught between Car and Rib	9	12	20	29	4	74
Caught between Car and Face	0	0	0	4	0	4
Caught between Car and Roof	5	9	4	7	2	25
Caught between Cars (not Coupling)	7	11	2	10	3	33
Runaway Car or Trip	3	10	7	11	0	31
Jumping on or off Car or Locomotive	2	3	8	11	0	24
Collisions	3	6	5	8	3	25
Derailments	7	6	13	15	2	43
Killed by Cars, not Stated	5	5	62	54	8	134
Roof Falls	1	0	1	0	8	10
Animals	1	7	8	7	0	23
Contact Trolley Wire	2	4	12	18	3	39
Miscellaneous	1	3	1	4	1	10
Total Haulage Fatalities	75	139	213	292	65	782
Total Coal-mining Fatalities	611	1122	856	1020	222	3831
Per Cent Haulage Fatalities	12.3	12.4	24.9	28.6	29.3	20.4

four years are selected because they represent recent average conditions and the statistics are complete. The average number of haulage employees per year was 12 493; hence the duties of mine haulage required more than one-seventh of the entire number of coal mine workers in the state. For these same years in Illinois there were 243 fatalities directly attributable to haulage as against 870 total coal-mine fatalities. These fatalities averaged respectively, 60.75 and 217.5 annually. Since 60.75 haulage fatalities were sustained among 88 274 employees, this was 1 for each 1453 men employed about coal mines.

Even among those 75 781 employees who positively had no duties connected with haulage, Table 19 shows that the annual haulage fatalities for this same four-year period averaged 19.25, or one in 3937, thus leaving an average of 41.5 haulage employees killed each year in the discharge of their duties. There being 12 493 such employees, it follows that the mortality was one per 301 men.

The number of deaths occurring year by year naturally increases with increase in production and number of men engaged. Thus, as shown in Table 19, while the production has increased rapidly, it has always been more than 1 000 000 tons of coal per fatal haulage accident, the best record being in 1905, slightly more than 3 000 000 tons, and the lowest in 1913, 1 124 476 tons. The average for the whole period is about 1 500 000 tons.

The graph, Fig. 28, shows a periodicity in the fatalities directly attributable to underground haulage. It can be seen that the peaks and depressions do not coincide with similar features of the curve for total coal-mining fatalities, Fig. 27. The large numbers of fatal

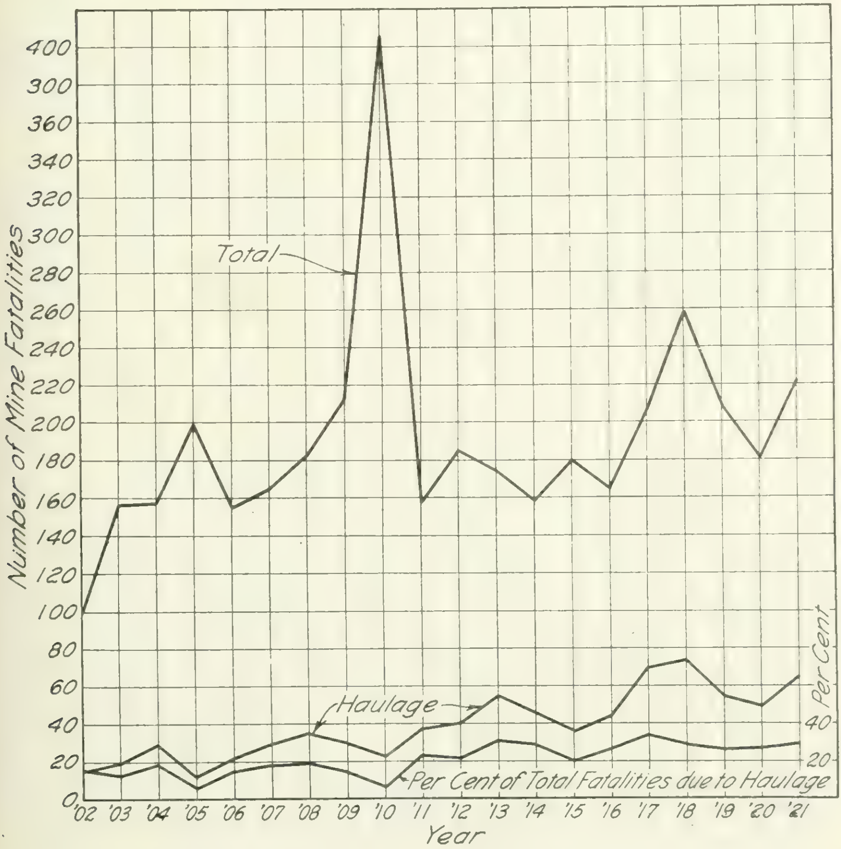

FIG. 27. GRAPH OF ILLINOIS COAL MINE FATALITIES

TABLE 19

RELATION BETWEEN COAL PRODUCTION AND HAULAGE FATALITIES IN ILLINOIS

Year	Production	Total Haulage Fatalities	Total Fatalities Haulage Employees	Fatalities Non-Haulage Employees	Fatalities to Drivers		Fatalities to Motormen and Trip Riders		Tons per Haulage Fatality
					No.	Per Cent	No.	Per Cent	
1902	30 021 300	15	13	2	10	66.70	..	.....	2 001 420
1903	34 955 400	19	15	4	14	73.11	..	.....	1 839 757
1904	37 077 897	29	21	8	16	55.17	..	.....	1 278 544
1905	37 183 374	12	9	3	8	66.66	..	.....	3 098 614
1906	38 317 581	22	18	4	15	68.18	1	4.54	1 741 708
1907	47 798 621	29	22	7	17	58.62	1	3.44	1 648 228
1908	49 272 452	35	31	4	25	71.42	..	.....	1 407 784
1909	49 163 710	30	23	7	18	60.00	2	6.66	1 638 790
1910	48 717 853	23	23	0	20	86.95	1	4.34	2 118 170
1911	50 165 099	37	30	7	22	59.45	3	8.10	1 355 813
1912	57 514 240	40	32	8	20	50.00	5	12.50	1 437 856
1913	61 846 204	55	37	18	22	40.00	6	10.90	1 124 476
1914	60 715 795	45	35	10	20	44.44	10	22.22	1 350 352
1915	57 601 694	36	25	11	13	36.11	9	25.00	1 600 047
1916	63 673 530	44	29	15	18	40.90	5	11.36	1 446 443
1917	78 983 527	70	48	22	28	40.00	14	20.00	1 128 336
1918	89 979 469	74	51	23	20	26.66	21	28.00	1 199 726
1919	75 099 784	55	36	19	14	25.00	16	28.57	1 341 067
1920	73 920 653	49	25	24	9	18.38	7	14.29	1 508 585
1921	80 121 948	65	54	11	18	27.69	28	43.08	1 232 645
Ave and Totals	1 122 130 131	784	577	207	347	44.25	129	16.45	1 431 287

accidents that occurred in 1905, 1910, and 1915 were caused by serious disasters such as fires and explosions but it would seem that haulage employees suffered least of all the classes of underground employees. The peaks in the curve for haulage fatalities, Fig. 28, preceded the peaks of total fatalities by a year or two in each instance and the question suggests itself, did not the haulage employees naturally become more careful after each time of heavy loss and in consequence conduct their duties with special attention to "safety first"?

The relative hazards incident to the occupations of those killed in connection with haulage are shown by Table 20 and the graph, Fig. 29. It can be expected that with the more extended use of mechanical haulage, with increased speed and size of equipment, and with the utilization of haulage ways as traveling ways, the hazard to employees other than haulage employees will be increased. During the period 1902 to 1921, inclusive, non-haulage employees sustained 28.8 per cent of the total haulage fatalities, as shown by Table 20.

Since 1915 the number of non-haulage employees killed each year has exceeded the number of motormen and trip riders killed; and, since 1918, has even exceeded the number of drivers or motormen and trip riders killed—the largest groups among haulage employees. The classes of employees included in haulage fatalities are given in Table 20 in which the first ten occupations are essentially connected with haulage. It seems that the accidents to non-haulage employees can be most readily prevented and should be given special attention.

Table 21 gives a comparison of the haulage hazard for various counties in the state for a period of nine years. As Franklin county not only has the largest number of fatalities per million tons, but is now the largest producing county in the state, a detailed study of the casualties in that county was made (Table 22). Of the mine fatalities

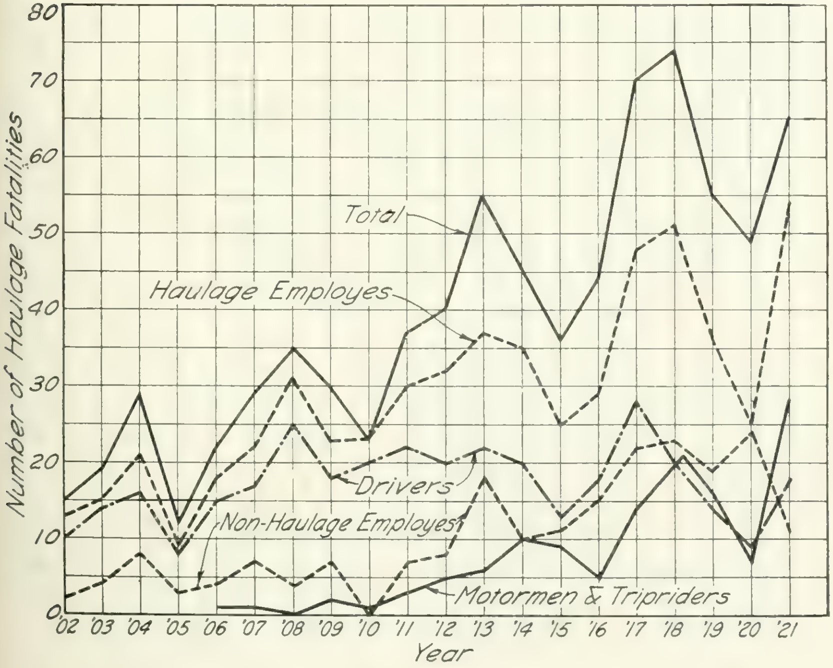

FIG. 28. GRAPH OF ILLINOIS COAL MINE HAULAGE FATALITIES

TABLE 20

HAULAGE FATALITIES IN ILLINOIS—CLASSIFIED BY OCCUPATIONS

Occupations	Period					
Haulage Employees:	1902–05	1906–10	1911–15	1916–20	1921	Totals
Drivers	48	95	97	89	18	347
Trip Riders	0	5	24	44	19	92
Trappers	6	7	11	16	5	45
Motormen	0	0	9	17	9	35
Track and Road Men	2	2	3	9	2	18
Spraggers	1	2	1	1	1	6
Switchmen	0	1	2	2	0	5
Couplers	0	1	1	3	0	5
Electricians	0	0	0	4	0	4
Grippers	0	1	0	0	0	1
Total Haulage Employees	57	114	148	185	54	558
Non-Haulage Employees:						
Miners	14	12	38	59	7	130
Laborers	2	8	11	17	1	39
Managers and Assistants	1	1	4	12	1	19
Cagers	0	3	9	5	1	18
Timbermen	0	1	2	4	1	8
Pipemen and Pumpmen	0	0	0	3	0	3
Machine Runners	0	0	0	2	0	2
Bratticemen	0	0	0	2	0	2
Blacksmiths	0	0	0	2	0	2
Hoist Engineers	1	0	0	0	0	1
Shot-firers	0	0	0	1	0	1
Mining Engineers	0	0	1	0	0	1
Total Non-Haulage, Employees	18	25	65	107	11	226
Total Fatalities	75	139	213	292	65	784
Per Cent, Haulage Employees	76.0	82.0	69.5	63.4	83.1	71.2
Per Cent, Non-Haulage Employees	24.0	18.0	30.5	36.6	16.9	28.8

in Franklin county during 15 recent years, 22 per cent have been due to haulage, while during the last five years 28 per cent have been due to the same cause. Undoubtedly, large producing mines, large capacity cars, and high speed are the chief reasons for the increased number of haulage fatalities.

Table 23 presents statistics for one year for the non-fatal accidents that occurred in a selected group of typical Illinois coal mines. In this same year, 1919, the total number was 2620, so that roughly speaking this table covers one-third of all such accidents in the state.

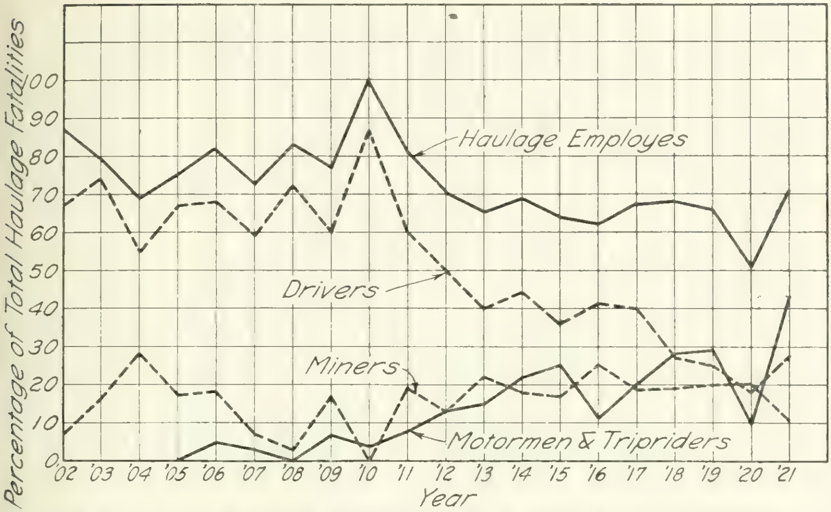

FIG. 29. GRAPH SHOWING PERCENTAGES OF FATALITIES BY OCCUPATIONS

TABLE 21

RELATION OF HAULAGE FATALITIES TO PRODUCTION
For Period of Nine Years in 19 Coal-Mining Counties of Illinois

County	Tonnage	Fatalities	Fatalities per Million Tons
Franklin	69 263 400	97	1.40
Montgomery	26 259 295	29	1.10
Jackson	6 948 458	7	1.01
Vermilion	26 908 926	24	0.90
Williamson	80 906 264	72	0.89
Saline	38 700 228	33	0.83
Fulton	19 595 117	14	0.71
Christian	18 901 802	13	0.70
Madison	35 766 010	25	0.69
Sangamon	53 303 653	36	0.68
Macoupin	49 194 248	33	0.67
Washington	4 483 649	3	0.67
Perry	19 182 399	12	0.62
La Salle-Bureau*	39 369 969	21	0.53
St. Clair	43 627 890	19	0.44
Peoria	9 773 729	4	0.41
Clinton	10 988 907	4	0.36
Randolph	9 103 527	1	0.11
Marion	9 453 052	0	0.00
Totals	570 730 523	447	0.78
Bal. of State	24 848 819	9	0.36
Total of State	595 579 342	456	0.76

*Northern Longwall Field

TABLE 22
FATALITIES IN COAL MINING, FRANKLIN COUNTY, ILLINOIS
For 17-year Period

Cause	1905	1906	1907	1908	1909	1910	1911	1912	1913	1914	1915	1916	1917	1918	1919	1920	1921	17-Year Period		Last 5 Years	
																		Total	Per Cent	Total	Per Cent
Falls, Coal, Rock	1	1	0	3	2	3	4	7	11	11	4	10	8	11	8	10	14	108	20.3	51	22.5
Cars, Locomotives	0	1	0	1	2	4	3	4	6	6	8	9	20	19	18	15	11	128	24.0	83	36.6
Explosions	50	0	0	4	43	3	0	0	1	1	54	9	2	21	24	0	2	214	40.2	49	21.6
Electricity	0	0	1	0	0	2	0	1	0	1	1	2	1	5	0	2	2	18	3.4	10	4.4
Miscellaneous	1	0	0	0	2	0	2	0	0	1	1	0	1	0	0	3	0	11	2.1	4	1.7
Shaft	0	3	0	2	0	1	0	0	5	1	0	1	0	8	3	2	1	27	5.1	14	6.2
Surface	0	0	0	1	0	0	0	0	0	0	2	2	5	0	2	4	3	18	3.4	14	6.2
Explosives	0	1	1	1	0	0	0	0	2	0	1	0	0	0	0	2	0	8	1.5	2	0.8
Totals	52	6	2	12	49	13	9	12	25	21	71	33	37	64	55	33	38	532	100.0	227	100.0

TABLE 23

NON-FATAL ACCIDENTS FOR GROUP OF ILLINOIS MINES, FOR YEAR 1919

Occupations	No. Injured	No. Shifts Lost
Haulage Employees:		
Motormen	44	1 011
Trip Riders	112	3 448
Trappers, Spraggers	15	301
Couplers	10	440
Trackmen	26	689
Electricians	20	352
Total Haulage Employees	227	6 241
Other Underground Employees	572	15 869
Surface Employees	71	1 485
Total Non-Fatal Accidents	870	23 595
Per Cent, Haulage Employees	26.1	26.4

Comparing injuries by the relative losses of time sustained by the victims, the average time lost per accident was 27.1 shifts, this applying to all occupations of coal mining, both underground and surface. Although the electricians' duties should properly be distributed between haulage, coal-cutting, and illumination, they are charged to haulage exclusively. On this basis, we have a total number of 227 injuries that caused a loss of 6241 shifts or approximately 27.5 shifts per accident. The significance of this analysis is that the injuries sustained from haulage appear to be about equal in severity to the average of all coal-mining non-fatal injuries. This of course has no direct bearing upon relative hazards nor upon fatalities. This appears in a different manner in the last line of Table 23 which shows that haulage employees sustained not only 26.1 per cent of the accidents but also 26.4 per cent of the lost time.

Table 24 is presented to compare coal mine haulage fatalities in Illinois with those in the bituminous district of Pennsylvania. The Pennsylvania data are from the Statistical Analysis of Coal Mine Accidents compiled by the Insurance Department of Pennsylvania. This table covers the 5-year period, 1916-1920, thus representing present conditions, and shows production tonnages for several classes of employees. Pennsylvania produced more than twice as much coal as did Illinois with about twice as many coal-mine employees. Various interesting comparisons may be noted in the column of Ratios.

TABLE 24

UNDERGROUND HAULAGE FATALITIES IN BITUMINOUS MINES OF PENNSYLVANIA AND ILLINOIS

5-Year Period, 1916-1920

Items	Pa.	Ill.	Ratios	Comments
Total Coal Production	831 877 000	381 656 963	2.18 : 1	
Total No. Men Employed	864 878	427 273	2.02 : 1	Pa. double Ill.
Tons Coal per Employee	961.8	893.2	1.07 : 1	Close
Haulage Fatalities:				
Total	557	292	1.91 : 1	Almost 2 : 1
To Non-Haulage Employees	221	103	2.14 : 1	Parallel with production
To all Haulage Employees	318	189	1.68 : 1	5 : 3
To Loco. Employees	209	61	3.43 : 1	Notable difference
To Mule Drivers	99	89	1.11 : 1	Nearly even
Percentage of Haulage Fatalities:				
Suffered by Haulage Employees	57.09	64.72	1 : 1.13	
Suffered by Loco. Employees	65.72	32.27	2.03 : 1	Pa. double Ill.
Suffered by Mule Drivers	31.13	47.09	1 : 1.51	Ill. 51 per cent greater
Tons of Coal Produced for each Fatality:				
Haulage	1 493 495	1 307 044	1.14 : 1	
Haulage Employee	2 615 965	2 019 349	1.29 : 1	
Loco. Employee	3 980 272	6 256 671	1 : 1.57	
Mule Driver	8 402 798	4 288 280	1.96 : 1	Pa. double Ill.
Non-Haulage Employees	3 764 149	3 705 407	1.01 : 1	Practically equal
No. of Employees for each Fatality:				
Due to Haulage	1552.7	1463.3	1.06 : 1	
To Haulage Employees	2719.7	2260.7	1.20 : 1	
To Non-Haulage Employes	2913.5	4148.3	1 : 1.42	
To Loco. Employees	4138.2	7004.5	1 : 1.69	
To Mule Drivers	8736.1	4800.8	1.82 : 1	

33. *Comparative Hazards in Locomotive and Animal Haulage.*—The question arises as to whether or not locomotive haulage is more dangerous than animal haulage. Analysis of Illinois statistics on this subject shows that a direct answer to the inquiry is imposible, but the statistics in the Annual Coal Report of Illinois for the year 1921 may be accepted as fairly representative of present-day conditions. In that year there were in the coal mines 2892 locomotive men, 4229 drivers, and 278 boss drivers. Of all classes of underground employees, numbering 81 708, 39 men were killed by locomotive haulage and

26 men by animal haulage, a total of 65 fatalities (see Table 19). Of these 65 fatalities, 54 were among the 23 453 employees connected with haulage. There were 7399 men employed in moving trips of coal, 2892 of these being motormen and trip riders and the remaining 4507, mule drivers. Of the 54 haulage-employee victims 8 were men other than trip men (5 trappers, 2 tracklayers, and 1 spragger), 28 were locomotive men, and 18 were drivers (see Table 20). Beside the 7399 trip men, there were 6054 haulage employees such as trappers, spraggers, trackmen, stablemen and electricians. (In the case of electricians it is assumed that about one-half their time is occupied with mining work connected with mining machines, illumination and pumping.) There were 19 haulage fatalities among the 74 309 employees other than trip men. Of these 17 were due to locomotive haulage and 2 to mule haulage. In 1921 there were 2.03 locomotive men per locomotive and 1.19 mules per driver.

From the above data several deductions are possible:

(a) Of the 4507 drivers and boss drivers 18 were killed in their occupation. This is a rate of 3.994 men per thousand.

(b) Of the 2892 locomotive men 28 met death in their duties, this being a rate of one man per 103 men or 9.682 men per thousand.

(c) Locomotive men were thus under a hazard 2.42 times greater than were mule drivers.

(d) Among all classes of underground employees, locomotive haulage, with its 39 fatalities, was but one and one-half times as dangerous as mule haulage with its 26 fatalities.

(e) Among the 74 309 employees other than trip men (motormen, trip riders, drivers) fatalities were 19, this being a rate of one death per 3911 men or 0.256 men per thousand. The risk assumed by such workmen appears reasonably small. In comparing the 17 locomotive haulage fatalities with the 2 fatalities due to mule haulage we run upon the striking fact that, to nearly 91 per cent of all underground employees, locomotive haulage is eight and one-half times as dangerous as mule haulage.

Such calculations and deductions, however—leading to the conclusion that locomotive haulage is much more dangerous than mule haulage—have been based upon the numbers of employees only, whereas recent practice refers vital coal-mine statistics to tonnage of production.

There is a slow but general lessening of mule haulage on main lines. The last statistics gathered on this point—those for the year 1921—show that mules hauled less than one-tenth as much coal over main lines as did locomotives. The superiority of locomotives over mules for main haulage became fully evident to Illinois coal operators about fifteen years ago. Mules handled their maximum annual tonnage on main roads in 1907. Since that year, there has been a general diminution of this mule haulage with a simultaneous increase in the annual tonnage handled on main roads by locomotives. Using data from Table 1 and Table 19 for the years 1908 to 1921 inclusive, we find that for a total of 674 766 930 tons of coal hauled by locomotives on main lines there were 334 fatalities and that for 185 986 960 tons hauled by mules there were 321 fatalities. This means that the respective tonnages per fatality were 2 020 260 and 579 398 and indicates that mule haulage is nearly 3.5 times as dangerous as locomotive haulage when computed from the standpoint of tonnage handled.

34. *Accident Prevention Measures.*—The safeguards or measures installed to prevent accidents are usually determined by their relative necessity. Generally speaking favorable natural conditions of haulage, with easy grades and good roadbed, standard haulage equipment in good repair, and strict enforcement of safe practices are prime requisites for safe haulage and efficient operation. There are dangers inherent to such acts as switching, spragging, coupling, jumping on and off cars and locomotives, and handling animals, but these risks may be minimized by strict adherence to and practice of safety-first principles. If these accidents are due to inadequate or poorly maintained equipment or to failure to inculcate safety principles among the employees, certain responsibilities must be assumed by mine owners. It is not, however, the purpose of this discussion to decide upon specific methods of minimizing the accident hazards that attend mine haulage but to study the occurrence and relative numbers of such hazards in the different occupations. When each operator duly analyzes the accidents that occur in his mine he will be in position to undertake corrective measures that will apply to his particular property. The frequent recurrence of accidents to a given group of workers emphasizes the need of accident-prevention regulations for that group. Statistics covering a period of years reveal the underlying

causes, and from them may be formulated more effective methods of combating the hazards.

For example, as the work of distributing and collecting cars underground must be maintained with a certain amount of speed, narrow haulageways with scant illumination are constant sources of danger. This hazard increases with the speed of haulage. In order to reduce this risk whitewash is applied to the walls and roofs of shaft bottoms in the larger mines of the state. The benefit of this treatment is especially marked in bottoms without concrete linings, but it is considerable even where concrete supports are erected. Mine superintendents believe that their men work more freely and cheerfully in the better illumination and that there are fewer accidents. Whitewash, moreover, possesses sanitary features that recommend its use in stables, first-aid rooms, offices, and waiting-rooms. Along main-haulage roads whitewash should be used in all manholes or refuges and upon all doors, as means of additional safety to employees. At partings and at all entry branchings in portions of mines remote from electrical lighting, whitewashed ribs and roofs greatly enhance the illumination and thus reduce accident hazards. There are several recipes for making whitewash which has the properties desired in underground use. The washes may be applied by either brushes or sprays—preferably the latter. Two or three coats should be applied with intermissions for due seasoning.

In all districts the personal factor is often the controlling element. It is generally agreed that such accidents as those due to falls, haulage, and handling explosives, have much in common and that mental and physical alertness and knowledge of the hazards are the essential safeguards. Workmen grow thoughtless of their own personal interests when continually subjected to dangers. It is very probable that the majority of the deaths classified by causes in Table 18 were due to carelessness of the victims themselves. The final responsibility is therefore placed to a very great extent on the individual worker.

35. *Safety Rules for Underground Haulage.*—Keep locomotives, cars and track equipment in good repair.

Standardize car equipment, such as bumpers and couplings.

Illuminate haulage ways so that men need not carry individual lights on motor tracks.

Have head-lights on locomotives and markers, gongs or lights on rear cars of trips.

Have safe clearance between cars and one or preferably both ribs of entry.

Maintain whitewashed refuge holes at regular intervals.

Use block fillers to top of rail-web in flangeways and wedge-spaces in frogs and switches.

Use low-voltage trolley current; support wire at short intervals, so that sag will not exceed 3 inches; guard trolley with boxing 3 inches lower than wires, especially where men travel, as at junctions and stations where man-trips are made up.

Start locomotives only on signal from trip riders and after giving warning bells. Ring bells before all junctions.

Keep car-doors and latches in repair and inspect reclosing.

Give special instructions for spragging and blocking cars.

Place limitations on speed of travel.

Maintain special instruction for motormen and trip riders regarding the making-up of trips.

Give instructions in coupling cars.

Impress on all working in the mine the necessity of personal caution.

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Bulletin 18. Tests on Clay Materials Available in Illinois Coal Mines, by R. T. Stull and R. K. Hursh. 1917. *Mailing weight, one pound.*

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Bulletin 22. Water-Gas Manufacture with Central District Bituminous Coals as Generator Fuel, by W. W. Odell and W. A. Dunkley. 1918. *Postage two cents.*

Bulletin 23. Mines Producing Low-Sulphur Coal in the Central District, by G. H Cady. 1919. *Postage two cents.*

Bulletin 24. Water-Gas Operating Methods with Central District Bituminous Coals as Generator Fuel, by W. A. Dunkley and W. W. Odell. 1919. *Postage two cents.*

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A STUDY OF EXPLOSIONS OF GASEOUS MIXTURES

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A STUDY OF EXPLOSIONS OF GASEOUS MIXTURES

I. General Statement Concerning Investigation

1. *Preliminary Statement.*—An investigation of explosions of gaseous mixtures at constant volume was undertaken by the Engineering Experiment Station in 1914, when C. R. Richards, Dean of the College of Engineering and Director of the Engineering Experiment Station—at that time Head of the Department of Mechanical Engineering—outlined an extensive series of experiments and defined the scope of the investigation which was to follow. Apparatus was designed and a few results were obtained showing the effect of the shape of the explosion vessel, and of turbulence at the time of firing, but it was found necessary to discontinue the work. In 1919 part of the apparatus was re-designed and work was resumed. This bulletin is a preliminary report of the methods pursued and the results thus far obtained.

2. *Scope of Investigation.*—The investigation as originally planned was to consist of experiments on the simple gases—namely, hydrogen, carbon-monoxide, methane, and possibly acetylene—to determine the effect of varying proportions of gas and air, initial pressure, shape of the explosion vessel, ratio of its volume to its area, position of ignition, and turbulence during the time of ignition, upon the rate of inflammation, maximum pressure, and time of explosion. The work was next to be extended to various combinations of gaseous fuels, including the commercial fuels used in practice for internal combustion engines, in order to determine the effect of different proportions of the various constituents upon the rate of inflammation, maximum pressure, and time of explosion.

Tests were also to be made with the temperature of ignition corresponding to the temperature found at the end of compression in different types of commercial internal combustion engines. It was expected that some method of heating could be developed that would preclude flameless combustion taking place at the surface of the heating element before the ignition temperature was attained. From this series conclusions might be drawn relative to the effect of variations in the carbon-hydrogen ratio upon factors leading to pre-ignition in internal combustion engines.

3. *Object of Investigation.*—The immediate object of the series of tests reported in this bulletin was a study of the physical phenomena involved in the explosions of various mixtures of illuminating gas and air. This includes the determination of the effect of different shaped explosion vessels, and of turbulence at the time of ignition, as well as a study of the heat loss from the burning mass of gas.

Any problems connected with the chemical changes involved have not been considered within the scope of the investigation.

4. *Acknowledgment.*—Acknowledgment is due to C. R. RICHARDS, Dean of the College of Engineering, for his constant interest and encouragement, his many helpful suggestions, and for the original outline defining the scope of the investigation.

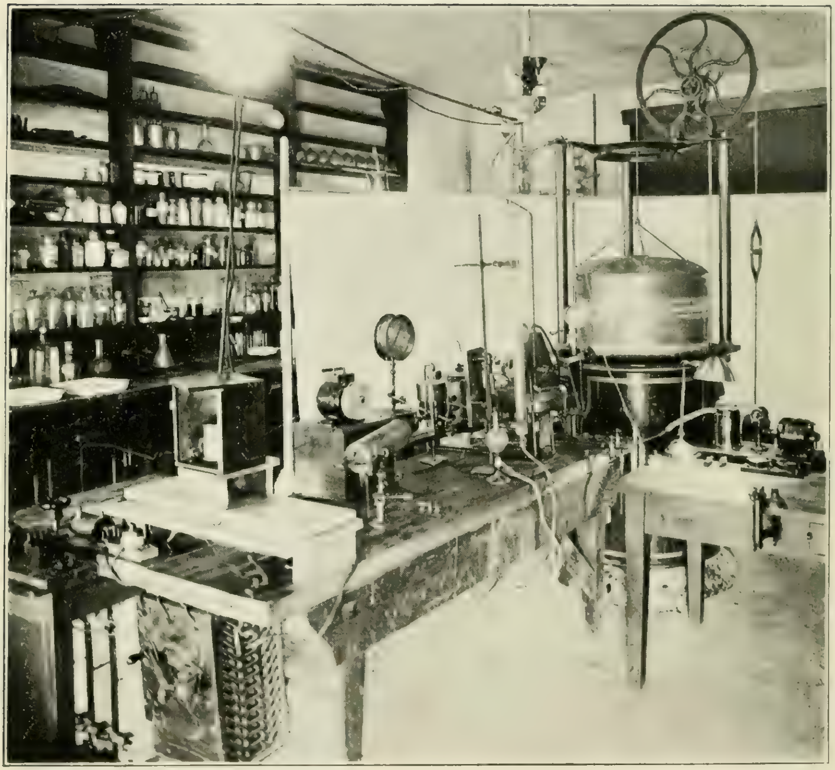
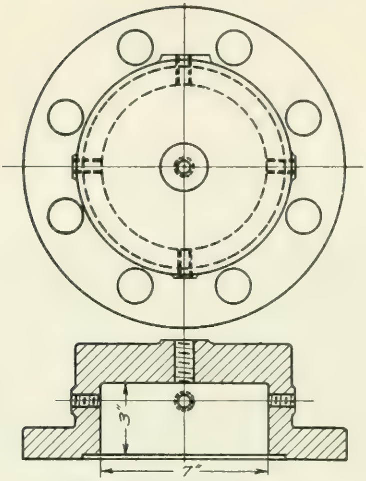


FIG. 1. GENERAL VIEW OF EXPLOSION APPARATUS

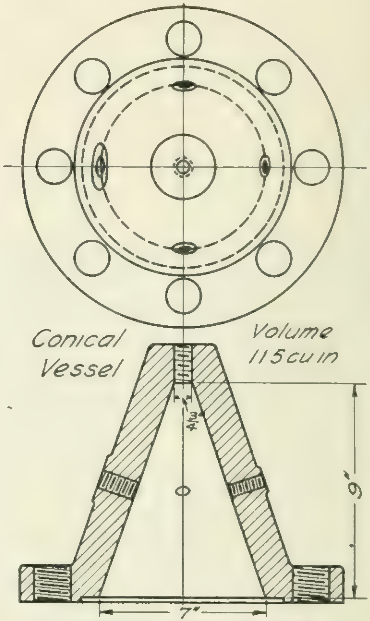
II. DESCRIPTION OF APPARATUS

5. *Explosion Apparatus.*—A general view of the apparatus used in this investigation is shown in Fig. 1. At the time the photograph was taken a conical explosion vessel was in use. The apparatus consisted of an explosion vessel, an indicator with a lamp and recording drum, a tuning fork, a gas holder, a small motor-driven vacuum pump, a measuring burette, a dead-weight tester for calibrating the indicator, and an induction coil. The photograph also shows the motors used for driving the fan in the explosion vessel and the recording drum for the indicator. This apparatus was essentially the same for all of the tests. At times it was found desirable to modify some of the details of the recording mechanism in order to facilitate the manipulation and to overcome vibration, but no changes in principle were made.

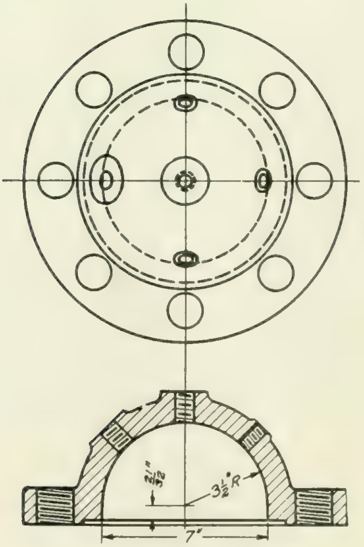
6. *Explosion Vessels.*—The explosion vessel in all cases consisted of a head of the desired shape bolted to a heavy base-plate, all parts being made of steel castings. Four different shaped heads were used, cylindrical, conical, hemispherical, and L-shaped, the last-named being similar to the combustion space of an L-head gasoline engine cylinder. The dimensions of these heads are shown in Fig. 2. They enclosed the same volume in all cases, and were designed to withstand explosion pressures of 2500 lb. per sq. in. They were centered accurately on the base-plate and were fastened to it by eight 1¼-in. cap screws. A thin paper gasket was used to prevent leakage. A 4-in. fan, having two blades and driven by a shaft passing through the center of the base-plate, afforded a means of agitating the mixture in the vessel. Threaded connections were provided in the walls of the vessels for the ignition plug and for gas piping. A diagram of the piping connections is shown in Fig. 3. Each vessel had two connections diametrically opposite for admitting and exhausting the air and gas. Heavy needle valves made from steel bars were used at these two points. All other valves were standard brass needle valves.



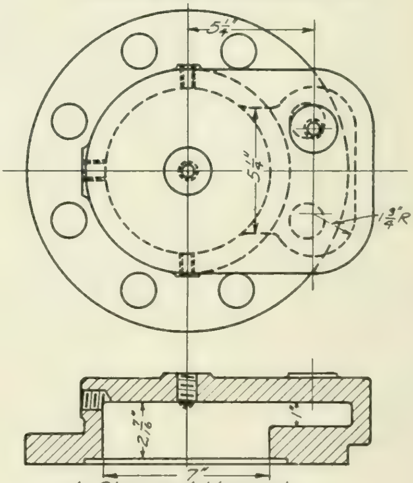
Cylindrical Vessel
Volume 115 cu. in. Superficial Area 143 sq. in.
Ratio-Area to Volume-1.238



Conical Vessel
Volume 115 cu in
Superficial Area-155 sq. in
Ratio-Area to Volume-1.342



Hemispherical Vessel
Vol-115 cu. in. Superficial Area-81 sq. in.
Ratio-Area to Volume-0.693



L-Shaped Vessel
Volume-115 cu. in. Superficial Area 187 sq. in.
Ratio-Area to Volume-1.618

FIG. 2. FORMS OF EXPLOSION VESSELS

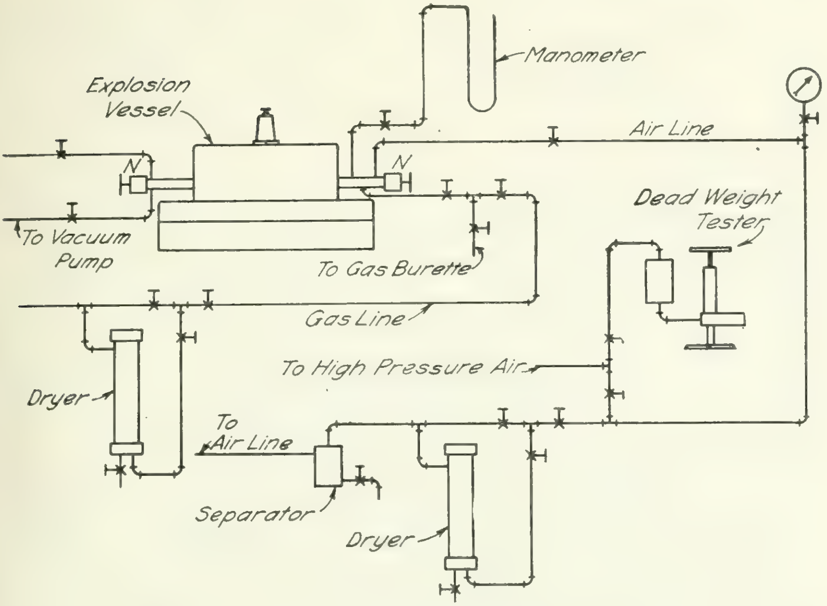


FIG. 3. GAS AND AIR PIPING

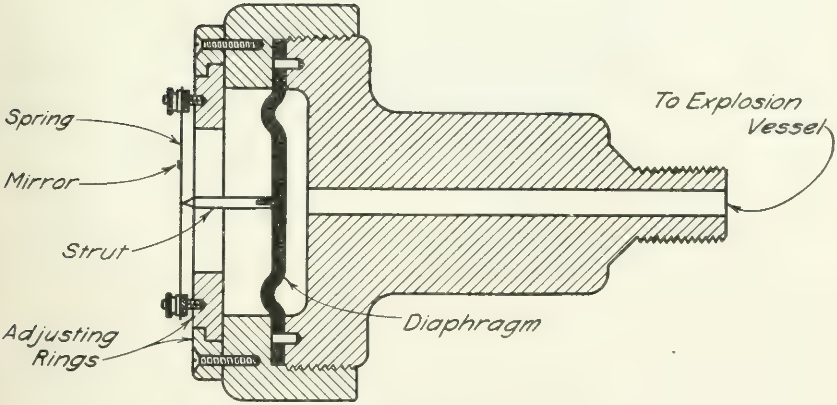


FIG. 4. INDICATOR DIAPHRAGM AND BODY

7. *Indicator.*—A diaphragm indicator and an optical recording mechanism was used for measuring the pressure in the vessel. The diaphragm and body for the indicator is shown in Fig. 4. The diaphragm was made of heat-treated chrome-vanadium steel, 3/64 in. thick, and had a semicircular corrugation concentric with the outside circumference of the disc. The purpose of the corrugation was to give greater flexibility and to prevent slipping in the fastenings. This arrangement also gives a straight line calibration for the indicator. A small threaded projection at the center of the diaphragm made it possible to connect the mirror system used for recording. The diaphragm was held in place by dowel pins and by the lock-ring, which was screwed down over the base as shown in Fig. 4. The body was then screwed into the cylinder and made metal to metal contact at the 45 degree shoulder, thus insuring a tight joint without the use of a gasket.

Fig. 5 shows the mounting for the mirror used to record the motion of the diaphragm. The base was formed by two concentric steel rings, the outer one of which was screwed to the lock-ring of the indicator. The inner ring was free to move around a central axis and could be locked in place by means of the four cap screws. The mirror was mounted on a steel spring extending between two studs. A small strut, screwed to the projection on the diaphragm, pressed against the spring at the center and communicated the motion of the diaphragm to the spring, which in turn tilted the mirror. This arrangement afforded a wide range of adjustment for the indicator mirror. The moving parts were very light and the effect of inertia was almost entirely eliminated. A calculation for the natural period of vibration of the diaphragm proved that the period was very short compared with the time of explosion and led to the conclusion that the instrument could follow the most rapid changes in pressure that were liable to occur.

The indicator was calibrated in place by means of a high-pressure dead-weight gage-tester, connected in the air line to the explosion vessel. A sample calibration curve is shown in Fig. 6, from which it is evident that the indicator retains its adjustment and calibration for long periods of time.

8. *Recording Apparatus.*—The recording mechanism consisted of a beam of light reflected from the concave mirror on the indicator, and focused on a strip of photographic paper carried on a revolving

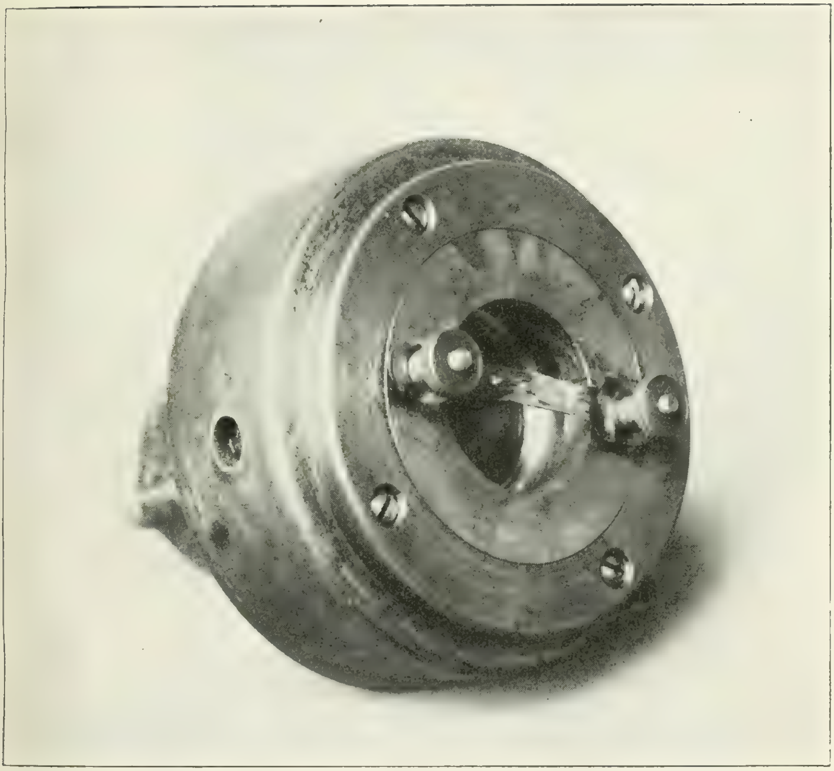


FIG. 5. INDICATOR WITH MIRROR ATTACHED

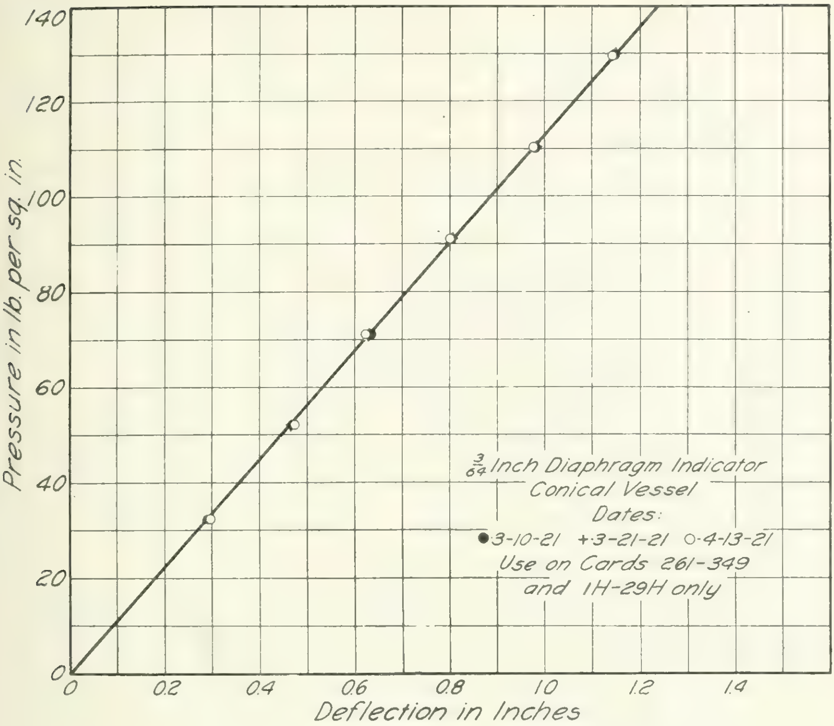


FIG. 6. CALIBRATION CURVE FOR OPTICAL INDICATOR

drum. The drum was 11 in. in circumference and was driven through a worm gear by a variable speed motor. A small enclosed arc lamp was used to project the beam of light on the mirror. A piece about 1/16 in. in diameter, cut from a concave mirror of 1 meter radius of curvature, was used and the arc light and drum were placed at such distances that a well defined image of the crater of the arc was obtained on the photographic paper. The records when developed showed a very narrow black line for the motion of the mirror. A special recording paper known as "Eastman Recording Paper No. 1" was used. This paper proved to be sufficiently rapid to give clear records, and was much more convenient to use than films. All records were developed immediately after being taken.

9. *Apparatus for Measuring Gas and Air.*—The illuminating gas used for the tests was taken from the city mains and stored in a gas bell having a capacity of 10 cu. ft. This amount was sufficient for several complete series of experiments, thus insuring gas of fairly constant composition.

On most of the tests the proper proportion of air and gas in the explosion vessel was obtained by making use of the partial pressures. The vessel was partly exhausted to some predetermined pressure. Gas was then admitted to the vessel until atmospheric pressure was again attained. The degree of exhaustion was measured by means of a mercury manometer inserted at the point shown in Fig. 3. A small rod bearing a platinum point was attached to a vernier head and the rod was inserted in one leg of the manometer. When the point touched the surface of the mercury it completed an electric circuit through the mercury and rang a bell. The vernier read directly to 0.01 in. and it was therefore possible to locate the surface of the mercury to this degree of precision. An accuracy of about 0.1 per cent was obtained in the air-gas ratio.

For the later series of tests the method of measuring the gas into the explosion vessel was changed. A 100-c.c. gas burette graduated to 0.1 c.c was connected into the gas line as shown in Fig. 3. The vessel was then partly exhausted and the gas was measured into it from the burette. This method proved to be more accurate and convenient than the first method described.

10. *Ignition Apparatus.*—The mixture in the explosion vessel was ignited by means of a spark passed across a 0.03 in. gap on a standard mica-insulated automobile spark-plug. A ¾-in. induction coil was used, drawing current from a 6-volt storage battery. The wiring diagram is shown in Fig 7. A contact attached to the circumference of the recording drum completed the primary circuit of the induction coil, and induced the spark at the gap in the spark-plug, thus igniting the mixture in the explosion vessel. A switch was provided in the primary circuit. The drum could then be started some time before the explosion was desired, and when it was revolving at constant speed, the switch could be closed and the explosion occurred when the contact was made on the recording drum. A small spark-gap in series with the plug was placed close to the sensitized paper on the revolving drum. A spark passed across this gap at the same

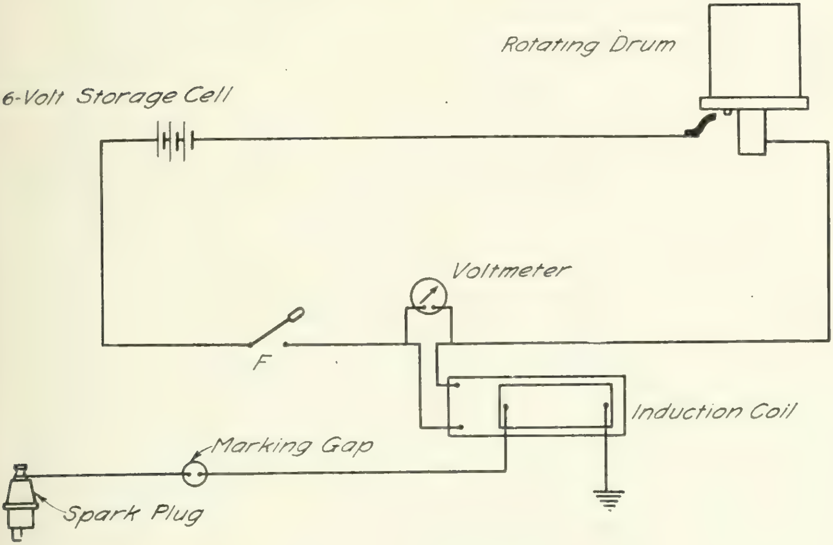


FIG. 7. WIRING DIAGRAM FOR IGNITION SYSTEM

instant the firing spark took place at the plug, and a dot was made on the record. This dot served as a means of locating on the record the exact time of firing. It should be noted that the time of firing did not coincide with the time at which the pressure in the vessel began to increase. The latter took place subsequent to the passing of the spark. When contact was made at the circumference of the drum, the circuit remained closed for about 0.1 sec., and from 6 to 10 separate discharges occurred across the spark-gap.

11. *Timing Apparatus.*—A tuning fork having a frequency of 100 vibrations per second was used in order to obtain the time of explosion. The tuning fork was electrically driven and had a small concave mirror mounted on one leg close to the base. A beam of light from the arc lamp was reflected from the mirror and focused on the recording paper placed on the revolving drum. When the fork was in vibration a sine curve was traced on the record, and the time could be computed by counting the peaks on the curve.

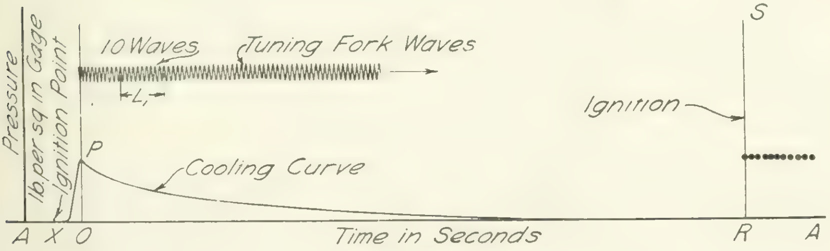
III. METHOD OF CONDUCTING TESTS

12. *Procedure in Making an Explosion.*—Before charging the explosion vessel with the mixture to be exploded, it was necessary to purge the vessel itself and the piping from any gas that remained from a previous test. Gas was first allowed to flow from the holder into the vessel in order to fill the line with fresh gas. The vessel was then thoroughly swept out with compressed air. This was done by admitting the air through the valve on one side, and allowing it to escape through the valve on the opposite side, during which time the fan inside of the vessel was kept running. The zero reading of the manometer was then observed, and the vernier bearing the platinum contact point was set in a position to give the required partial pressure for the gas. All valves except the one in the line from the vacuum pump were then closed, and the vessel was exhausted until the mercury in the open leg of the manometer fell below the contact point. Air was then allowed to enter the vessel very slowly until the bell in the manometer circuit gave warning that the surface of the mercury had just touched the contact point. When this was the case, the vacuum within the vessel was equal to the required partial pressure for the gas. The air valve was then closed, the gas valve was opened, and gas was allowed to enter until the pressure in the vessel was equal to that of the atmosphere. The valves, *N*, Fig. 3, were then closed.

In the later tests, where the measuring burette was used, the procedure outlined was somewhat modified. After purging the vessel it was exhausted to a pressure of about 10 in. of mercury absolute. The air valve was closed, and the required volume of gas was measured in from the burette. The exhaust valve was then opened, and air was allowed to enter the vessel until the pressure rose to that of the atmosphere. The valves, *N*, were then closed. The fan was run at about 300 rev. per min. during admission of the gas and was left running for a time considered sufficient to insure thorough mixing of the gas and air. If turbulence was desired the fan was allowed to run while the explosion was made.

The room was darkened, the sensitized paper was placed on the drum and the driving motor for the drum was started. The mirror on the tuning fork was covered and the arc light put in operation. The shutter was then opened and the zero or atmospheric line for the indicator was traced on the record, after which the shutter was closed. The mirror on the tuning fork was next uncovered, the fork set in vibration, the shutter on the drum opened again, and the firing switch *F*, Fig. 7, closed. When the contact at the circumference of the drum came into position the charge was automatically fired. The paper was then removed from the drum and placed in the developer.

13. *Calculation of Maximum Pressure and Time of Explosion.*—Typical records are shown in Figs. 8 and 26. The line *A-A*, Fig. 8, is the zero or atmospheric pressure line which was traced before the explosion was made: the serrated line represents the record of the tuning fork; and the line with the single peak is the record of the explosion and the subsequent cooling of the mixture. The first dot of the dotted line indicates the time at which the spark took place. *OP* is a line drawn perpendicular to the zero line and passing through the highest point on the pressure curve. In order to obtain the ignition point *X*, on the pressure curve *RS*, a perpendicular from the base line, is erected through the first dot on the spark curve. The distance between the position of the auxiliary spark gap, placed close to the paper, and the line representing the vertical travel of the spot of light from the indicator is then measured back from the foot of the perpendicular at *R*. This distance was determined by taking a sep-



Card No. 322
Initial Pressure-Atmospheric Air Gas Ratio-6
Maximum Pressure-82.5 lb. per sq. in Initial Temperature-75°F
Time of Explosion-0.054 Second

FIG. 8. TYPICAL RECORD OF EXPLOSION

arate record obtained while passing the spark with no explosive mixture in the vessel and with the drum at rest. For any given adjustment of the indicator the distance was constant. The horizontal distance, *XO,* is a measure of the time of explosion.

The distance XO is translated into seconds as follows. Let

N_1 = any arbitrary number of vibrations of the tuning fork, usually chosen as 10;

N_2 = the number of vibrations of the fork per second;

N = the number of vibrations contained in the distance $L = XO$;

L = the distance XO measured from the record;

L_1 = the horizontal distance covered by N_1 vibrations of the fork (Fig. 8);

T = time of explosion in seconds;

$$T = \frac{N}{N_2} \qquad (1)$$

$$\frac{L}{L_1} = \frac{N}{N_1}, \text{ or } N = \frac{LN_1}{L_1} \qquad (2)$$

$$T = \frac{LN_1}{L_1 N_2} \qquad (3)$$

Since $L = XO$, N_1 was usually chosen as 10, and the fork when calibrated was found to average 98 vibrations per second, equation (3) becomes:

$$T = \frac{0.102\ (XO)}{L_1}$$

IV. RESULTS

14. *Phenomena Concerned in Gaseous Explosions.*—If a homogeneous mixture of any inflammable gas and air at atmospheric temperature and pressure is enclosed in a vessel, the walls of which are also at atmospheric temperature, and the mixture ignited, inflammation will take place at a rate depending on the nature of the mixture. The heat developed by the combustion will increase the temperature of the gases, thus developing pressure within the vessel, but at the same time a loss of heat will take place from the gases to the walls of the vessel, thus tending to lower the temperature of the mixture and the resultant pressure. At some time during the process of combustion the loss of heat to the walls will balance the additional heat being evolved by the combustion, and the pressure will cease to rise. If a continuous record is made of the pressure in the vessel the curve will attain a maximum at this point, designated hereafter as the ''maximum pressure.'' The maximum pressure finally attained will not be affected by the time taken to complete the combustion other than by the amount of cooling during this period, but will be a function of the completeness of the combustion at the time that thermal equilibrium has been established. After maximum pressure has been reached the gas continues to cool by losing heat to the walls of the vessel, and the pressure decreases. If the cooling is allowed to continue indefinitely the gas and the vessel will ultimately cool to atmospheric temperature and if no contraction due to chemical combination has taken place, the pressure in the vessel will again be atmospheric.

The time elapsing from the instant of ignition to that of attaining maximum pressure will be designated as the ''time of explosion.'' This time is not as much a function of the energy liberated by the chemical reaction as it is of mechanical conditions within the mixture tending to bring the different parts of the mixture into more intimate contact, thus facilitating the communication of the flame from any one part to those adjacent.

Both maximum temperature and time of explosion will be influenced by the ''air-gas ratio,'' or the ratio by volume of air to gas in the original charge and the shape of the vessel within which the explosion takes place.

15. *General Statement Concerning Gas Used.*—The gas used in most of the tests was illuminating gas taken from the city mains. It was stored in a water sealed gas holder, and therefore was saturated with water vapor.

The earlier experiments on the cylindrical and those on the L-shaped vessel were made in 1915, while the later ones on the cylindrical, hemispherical, and conical vessel, were made in 1921. The calorific value and the hydrogen content of the gas was less in 1921 than it was in 1915. It was found by comparing the results obtained on the cylindrical head at the two different times that a correction could be made for the difference in calorific value of the gas. All the results of the earlier series were therefore corrected to bring them into conformity with the later series.

A comparison between the gas used in 1915 and that used in 1921 is given in Table 1.

16. *Designation for Tests.*—The tests have been grouped into a number of series in order to render them more convenient for the

TABLE 1

ANALYSES OF ILLUMINATING GAS

Constituent	Year 1915	Year 1921
CO_2 per cent	3.7	7.2
O_2 per cent	0.6	0.7
CO per cent	20.9	13.7
CH_4 per cent	28.8	23.6
C_2H_4 per cent	5.5	7.8
C_6H_6 per cent	2.8	2.7
H_2 per cent	18.7	11.7
N_2 per cent	19.0	32.6
*Calorific value in B. t. u.	562	504

*The calorific value is the lower calorific value based on one cu. ft. of gas at 62 deg. F. and 29.92 in. Hg.

purpose of discussion. Each series had for its object the determination of the effect of some one factor, as turbulence, conditions of ignition, or shape of explosion vessel, upon the maximum pressure and time of explosion, and consisted of a number of explosions made with different mixtures of gas and air ranging from the richest to the leanest mixture that could be exploded. No attempt was made to determine the exact limits of inflammability of the gas, but the mixtures were varied by half units in the scale of air-gas ratios until one was found that failed to explode. Some explosions of hydrogen and air were also made in the conical vessel in order to compare the experimental values of the maximum pressure with those obtained from theoretical considerations. Table 2 gives a schedule of the various series.

17. *Preliminary Tests.*—The curves obtained from Series 1 are shown in Fig. 9. This series was run on the cylindrical vessel,

TABLE 2

SCHEDULE OF TESTS

Series No.	Vessel No.	Shape of Vessel	Turbulence
1	1	Cylindrical	Without
2	2	L-shaped	Without
3	2	L-shaped	With
4	2	L-shaped	Without
5	2	L-shaped	With
6	2	L-shaped	Without
7	2	L-shaped	With
8	1	Cylindrical	Without
9	1	Cylindrical	With
10	3	Conical	Without
11	3	Conical	With
12	3	Conical	Without
13	3	Conical	With
14	4	Hemispherical	Without

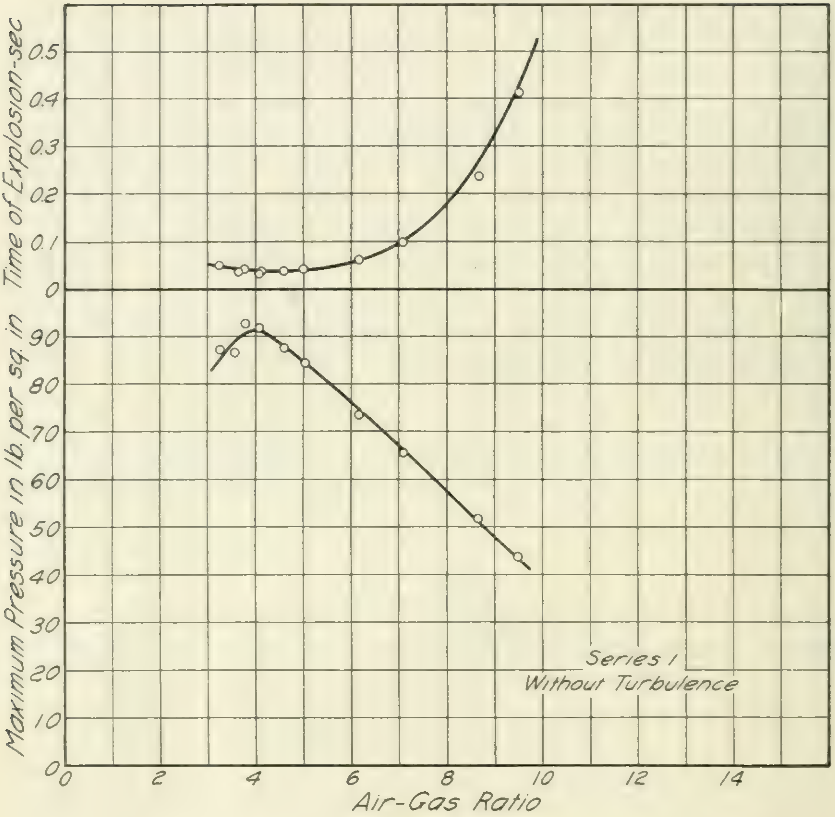


FIG. 9. EXPLOSION CURVES FOR CYLINDRICAL VESSEL

with ignition occurring flush with the upper surface at the center and without turbulence. The gas was admitted without having the fan running and the mixture was exploded after being allowed to stand for about 10 min. These curves are typical of all the explosion curves obtained. The series was run in 1915 with gas of a higher heating value than that used later, and the curves are given here because reference will be made to them in the study of the effect of the shape of the vessel upon the maximum pressure and time of explosion.

The maximum pressure occurred with an air-gas ratio of 3.9 to 1. The theoretical value of the air-gas ratio for maximum pressure is about 6 to 1. The discrepancy between these two figures may be explained by the fact that the 6 to 1 mixture contains a relatively larger amount of nitrogen than the stronger mixture, and the heat used to raise the temperature of the nitrogen reduces that which would otherwise be available for causing an increase of pressure.

The maximum pressure of 92.5 lb. per sq. in. and minimum time of explosion of 0.04 sec. are in close agreement with values of 91.0 lb. per sq. in. and 0.05 sec. which Dugald Clerk found for mixtures of Oldham gas and air.

18. *Turbulence during Ignition.*—The effect of turbulence at time of ignition for the L-shaped vessel is shown in the explosion curves of Figs. 10, 11, and 12. All the results for the L-shaped vessel were obtained with the same gas as that used for Series 1 and they are therefore strictly comparable. For Series 2 and 3, Fig. 10, the charge was fired flush with the upper surface and at the center of the main chamber. In Series 2 no stirring occurred at the time of firing, while in Series 3 the mixture was agitated by means of the fan during the time that the explosion took place. Series 4 and 5, Fig. 11, were run under the same conditions as Series 2 and 3 except that the charge was fired at the center of the valve chamber and flush with the upper surface of the vessel. In Series 6 and 7, Fig. 12, ignition occurred simultaneously in the main chamber and the valve chamber. Series 8 and 9, Fig. 13, were made using the cylindrical vessel, with ignition occurring flush with the upper surface and at the center of the head. These tests were made in 1921 and the gas used had a lower calorific value than that used for Series 1. For Series 10, 11, 12, and 13 the conical explosion vessel was used. In Series 10 and 11, Fig. 14, ignition took place at the vertex of the cone and in Series 12 and 13, Fig. 15, at a point 3 in. below the vertex.

In all cases the effect of turbulence at the time of ignition was very marked. The maximum pressure was increased and the time of explosion decreased for any given air-gas ratio. The effect was much more marked with the leaner mixtures than with the richer ones, indicating that with the lean mixtures combustion is retarded by the less intimate contact of the molecules. Turbulence also seemed to have a greater influence on the time of explosion, expressed as a

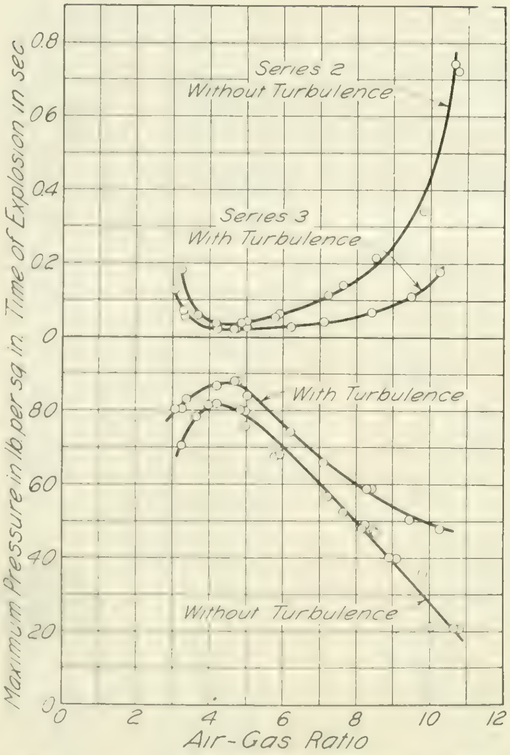


FIG. 10. EXPLOSION CURVES SHOWING EFFECT OF TURBULENCE FOR L-SHAPED VESSEL WITH IGNITION AT CENTER OF HEAD

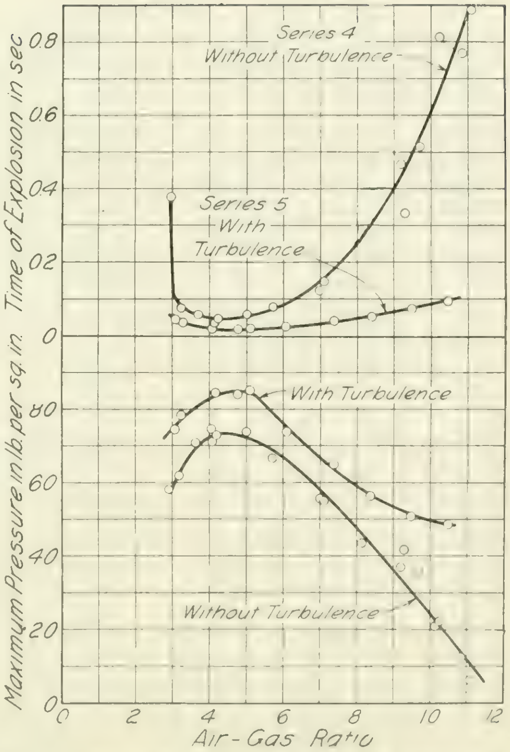


FIG. 11. EXPLOSION CURVES SHOWING EFFECT OF TURBULENCE FOR L-SHAPED VESSEL WITH IGNITION IN VALVE CHAMBER

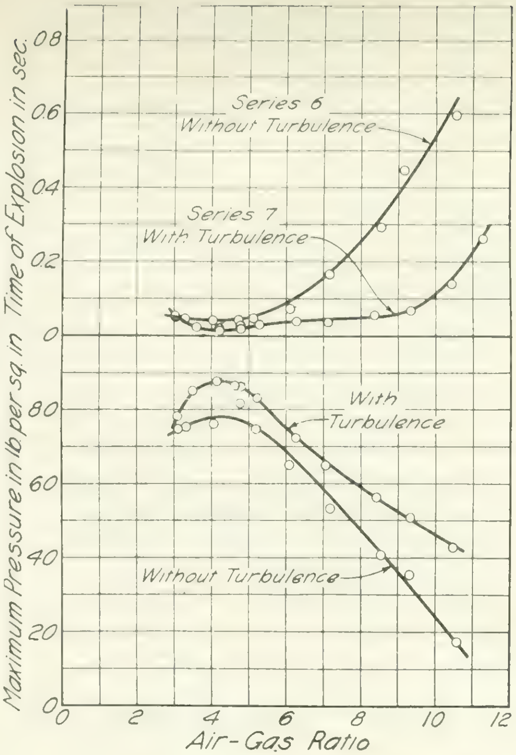


FIG. 12. EXPLOSION CURVES SHOWING EFFECT OF TURBULENCE FOR L-SHAPED VESSEL WITH IGNITION AT CENTER OF HEAD AND IN VALVE CHAMBER

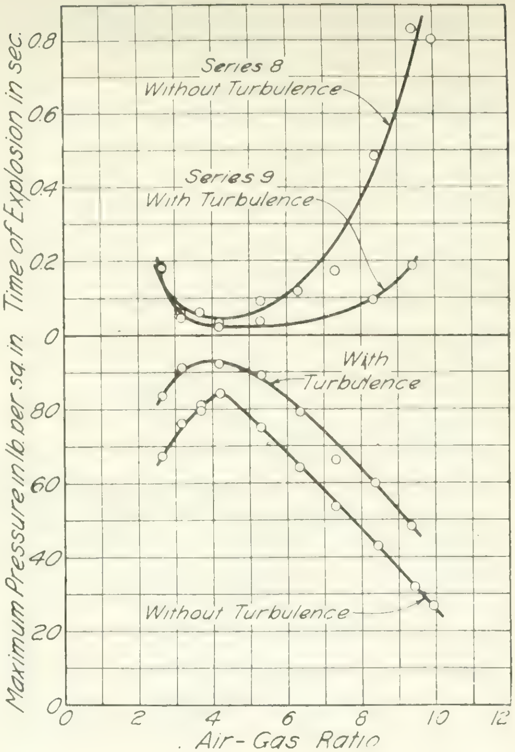


FIG. 13. EXPLOSION CURVES SHOWING EFFECT OF TURBULENCE FOR CYLINDRICAL VESSEL WITH IGNITION AT CENTER OF HEAD

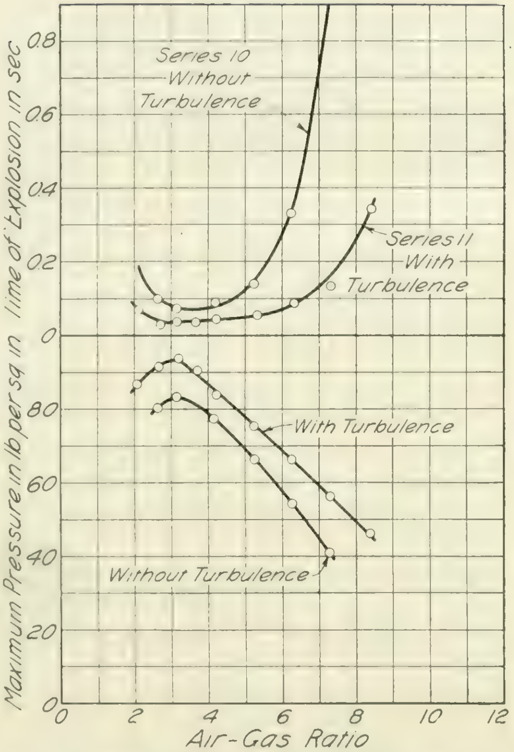


FIG. 14. EXPLOSION CURVES SHOWING EFFECT OF TURBULENCE FOR CONICAL VESSEL WITH IGNITION AT VERTEX

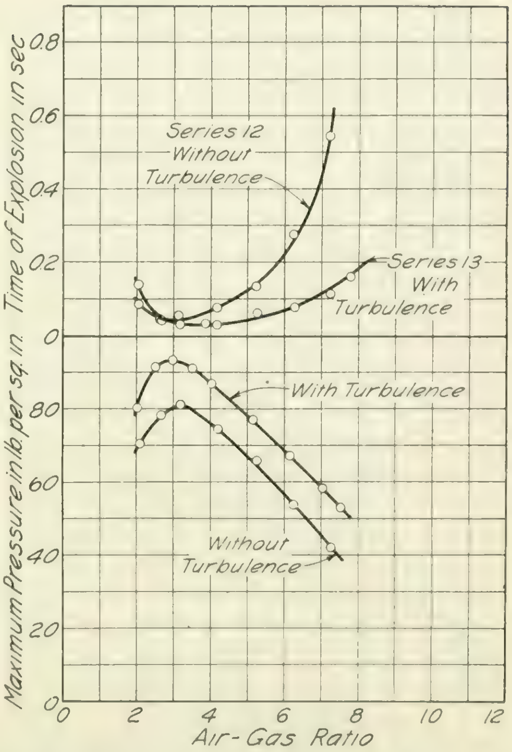


FIG. 15. EXPLOSION CURVES SHOWING EFFECT OF TURBULENCE FOR CONICAL VESSEL WITH IGNITION 3 IN. FROM VERTEX

percentage of the original time, than upon the maximum pressure. An explanation for this is given in Section IV, page 25. Besides increasing the maximum pressure for any given air-gas ratio, turbulence has the effect of increasing the value of the air-gas ratio at which the

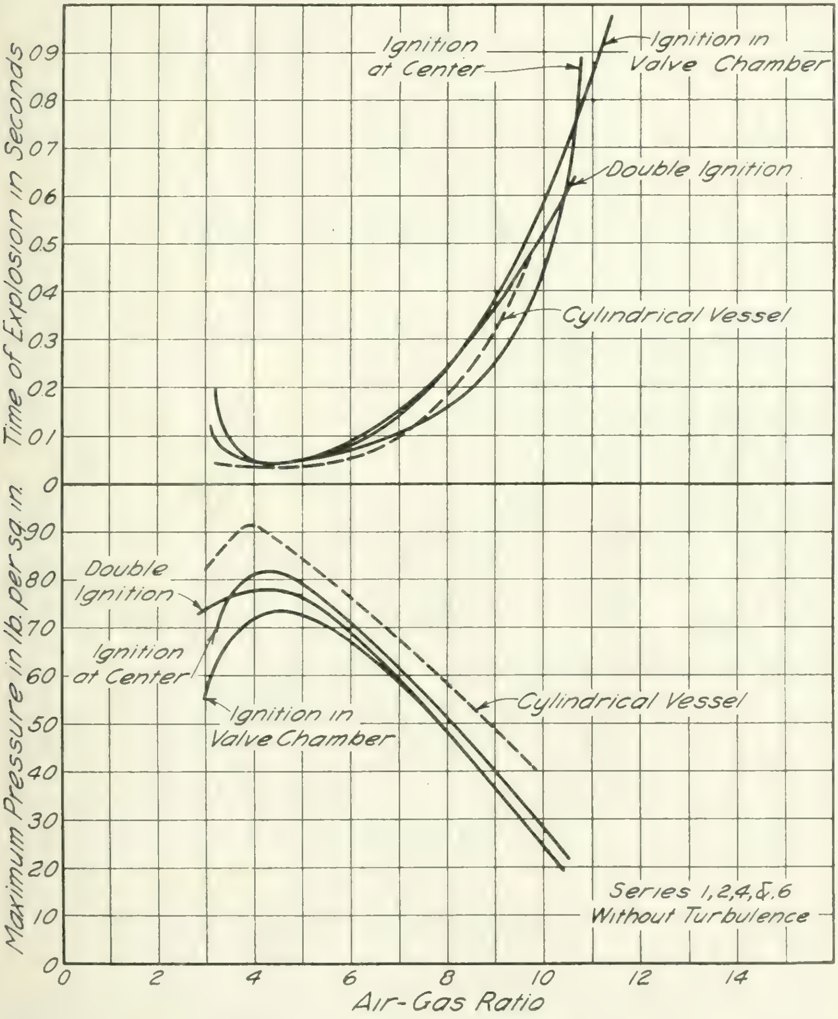


Fig. 16. Explosion Curves Showing Effect of Position of Ignition for L-Shaped Vessel

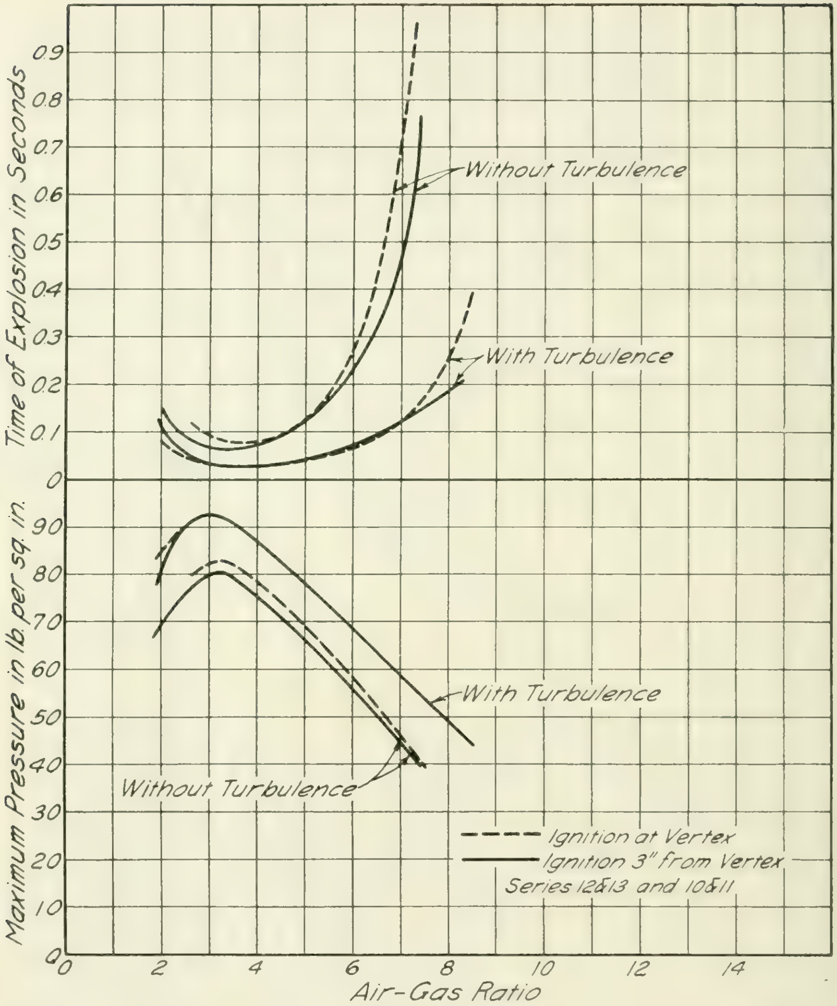


FIG. 17. EXPLOSION CURVES SHOWING EFFECT OF POSITION OF IGNITION FOR CONICAL VESSEL

maximum value of the maximum pressure occurs; that is, the best mixture corresponds to a leaner mixture when turbulence occurs than it does when no turbulence is present at the time of ignition.

19. *Position of Ignition Point.*—In Fig. 16 the curves resulting from Series 2, 4, and 6 have been superimposed in order to study the effect of the position of ignition. The curve for Series 1 has also been added and will be referred to in the discussion of the effect of the shape of the vessel. These curves show that the best position for ignition in an L-shaped combustion chamber is at the center of the head. When ignition occurs in a pocket the flame is cooled and is not propagated through the mixture as rapidly as when ignition takes place in the main chamber. Therefore, the slow burning mixture cools more during the explosion and the maximum pressure produced is not as great as it otherwise would have been.

In Fig. 17 the curves from Series 10, 11, 12, and 13 have been superimposed. These curves show that, when the fan in the vessel was not running, ignition at the vertex of the cone produced higher maximum pressures for all air-gas ratios than when ignition occurred 3 in. below the vertex of the cone. With turbulence, however, there seemed to be no difference in the maximum pressures produced with either position of ignition. This inconsistency may possibly be explained by the supposition that when ignition occurred at the vertex a pressure wave was started which advanced smoothly down the cone, expanding as though passing through a nozzle. When the gas was ignited further down within the cone, inflammation took place in all directions from the point of ignition, thus interfering with the formation of such a wave. Therefore, in the cases where no initial turbulence existed, the maximum pressure was higher when the mixture was fired at the vertex than it was when fired 3 in. below the vertex. When initial turbulence existed, however, the wave was broken up and the character of the combustion was the same for both methods of firing, as indicated by the coincidence of the two curves.

20. *Shape of Explosion Vessel.*—The explosion curves for Series 14 are shown in Fig. 18. This series was run on the hemispherical vessel with ignition occurring at the top of the vessel, and without turbulence.

In Fig. 16 the curve for the results from Series 1 for the cylindrical vessel was superimposed on those for Series 2, 4, and 6 for the L-shaped vessel. These series of tests were all made with the same gas in 1915 and are therefore strictly comparable. From this it may be noted that the maximum pressures for all air-gas ratios for the

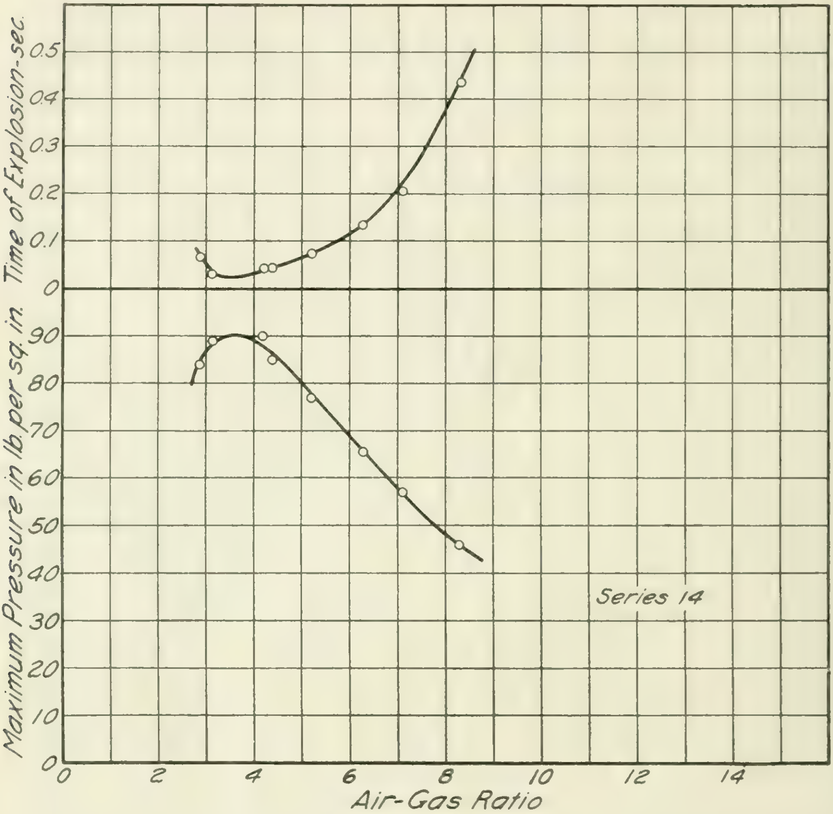


FIG. 18. EXPLOSION CURVES FOR HEMISPHERICAL VESSEL WITH IGNITION AT TOP AND WITHOUT TURBULENCE

cylindrical head were uniformly greater than those for the L-shaped head, even for the best condition of firing.

In Fig. 19 the curves for the results of Series 2, 8, 10, and 14 have been superimposed, in order to study the effect of the shape of the vessel upon the maximum pressure. Since Series 2 for the L-shaped vessel was run with a gas of different calorific value than that used for Series 8, 10, and 14, it was necessary to correct the curve for Series 2 in order to make all the curves comparable. This was done by comparing the curves of Series 1 and Series 8, which were run in the same vessel with the two different gases, one in 1915 and the other in 1921. These curves were found to have exactly the same shape, while the curve for Series 1 was practically 10 per cent higher than that for Series 8. This difference in the curves was attributed partly to the fact that the calorific value of the gas was higher for Series 1 than for Series 8. The hydrogen content of the gas was also greater in the former case. The rate of inflammation would be greater and consequently the time of explosion would be less; hence the amount of cooling before the attainment of maximum pressure would be less. The maximum pressures obtained for the same vessel from the gas used in 1915 would therefore be higher than those obtained from the gas used in 1921. The curve for Series 2 for the L-shaped vessel run in 1915 has therefore been lowered by 10 per cent, and has been plotted as the equivalent curve for the L-shaped vessel in Fig. 19.

From Fig. 19 it is evident that the hemispherical shape is best for the explosion vessel. The conical and cylindrical vessels seem to be equally good, and both have a marked advantage over the L-shaped vessel. Since the same volume was enclosed by all of the vessels used, it is probable that the differences in maximum pressure observed are

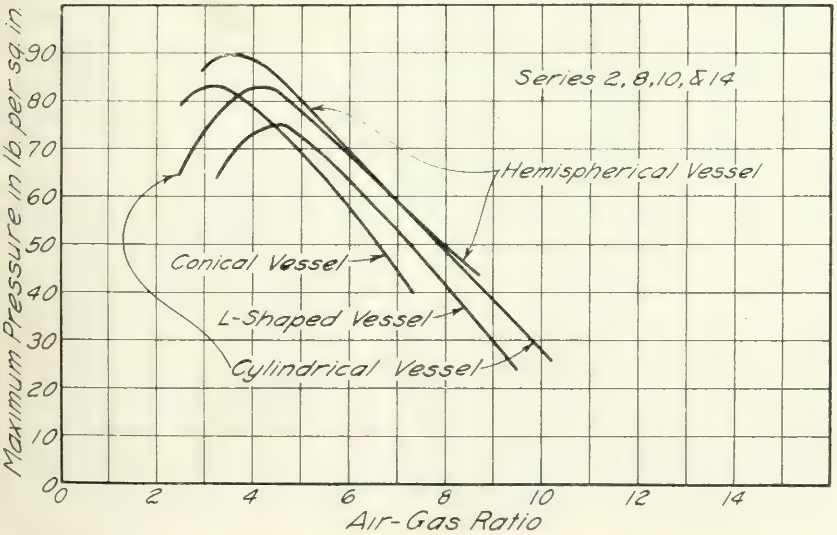


FIG. 19. MAXIMUM PRESSURE CURVES SHOWING EFFECT OF SHAPE OF EXPLOSION VESSEL

due to variation in the amount of surface exposed to the hot gases, which in turn influences the heat loss during the explosion. It is also possible that in the case of the L-shaped vessel complete combustion does not occur when the mixture is fired in the main chamber, since the flame must be cooled by passing into the narrow opening of the valve chamber.

Table 3 gives the ratio of the area of the inside surface to the volume for each vessel, and the greatest maximum pressures, taken from Fig. 19.

TABLE 3

RELATION BETWEEN MAXIMUM PRESSURE AND RATIO OF AREA TO VOLUME OF VESSEL

Shape of Vessel	Ratio A/V	Maximum Pressure
Hemispherical	0.69	90.0
Cylindrical	1.24	84.0
Conical	1.34	83.0
L-shaped	1.62	75.0

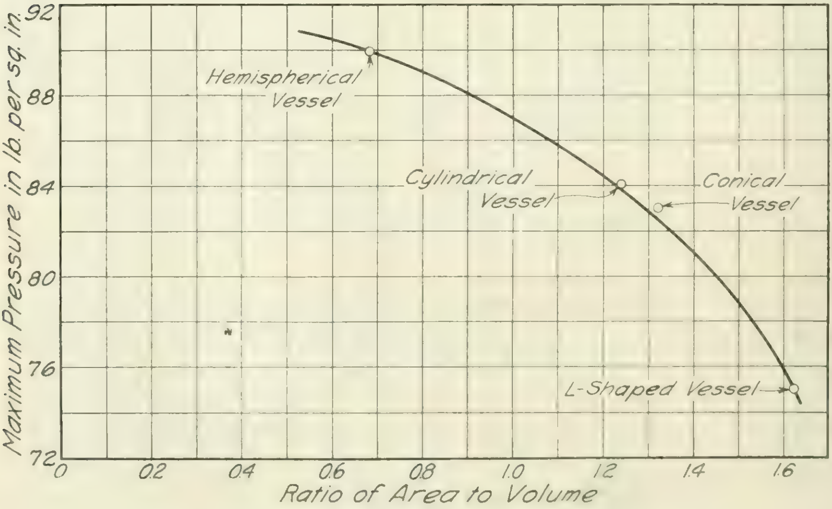


FIG. 20. CURVE SHOWING RELATION BETWEEN MAXIMUM PRESSURE AND RATIO OF AREA TO VOLUME OF VESSEL

The relation between maximum explosion pressure and the ratio of area to volume is shown in Fig. 20, which indicates that the cooling effect increases as the inside area increases.

21. *Rate of Cooling after Explosion.*—After the maximum pressure has been attained the enclosed gases in the explosion vessel start to cool. The cooling then continues at a rate which, if the combustion is complete and no after-burning takes place, is a function of (1) the air-gas ratio, (2) the character of the inside surface of the walls of the vessel, and (3) the ratio of inside wall surface to volume of the vessel. The curves in Fig. 21 show the rate of cooling for the products of combustion remaining in each of the vessels used, after the explosion of a mixture of 3 parts of air to 1 part of gas by volume. The initial temperature was chosen in each case as 2912 deg. F., and the curves were then constructed from the cooling curves on the indicator diagrams after allowance had been made for the contraction in volume due to the chemical reactions involved.

In Fig. 22 the drop in temperature from the initial value of 2912 deg. F. in 0.2 sec. and 0.5 sec. respectively after the time at which the gas was at the temperature of 2912 deg. F. has been plotted against the ratio of the area to the volume of the vessel. Since the volume was the same for all of the vessels, these curves indicate that the rate of cooling increases as the inside wall surface increases. If the nature of the inside surface is the same, it appears that the rate of cooling is a function of the area of the inside walls. The inside surface of the conical vessel was more polished and cleaner than the same surfaces for the other vessels. More radiant heat was therefore reflected from the walls of the conical vessel than from the walls of the others and the mixture cooled more slowly in the conical vessel. This may serve as an explanation for the fact that the point representing the drop in temperature in this vessel fell below the curve.

22. *Explosions of Hydrogen and Air.*—A mixture of one part of hydrogen to 2.4 parts of air by volume was exploded in the conical vessel in order to determine the relation between the maximum pressure and the time of explosion. This mixture contained the theoretical amount of oxygen necessary for the complete combustion of the hydrogen.

$$H_2 + \tfrac{1}{2}\,O_2 + 1.9\,N_2 \leftrightarrows H_2O + 1.9\,N_2$$

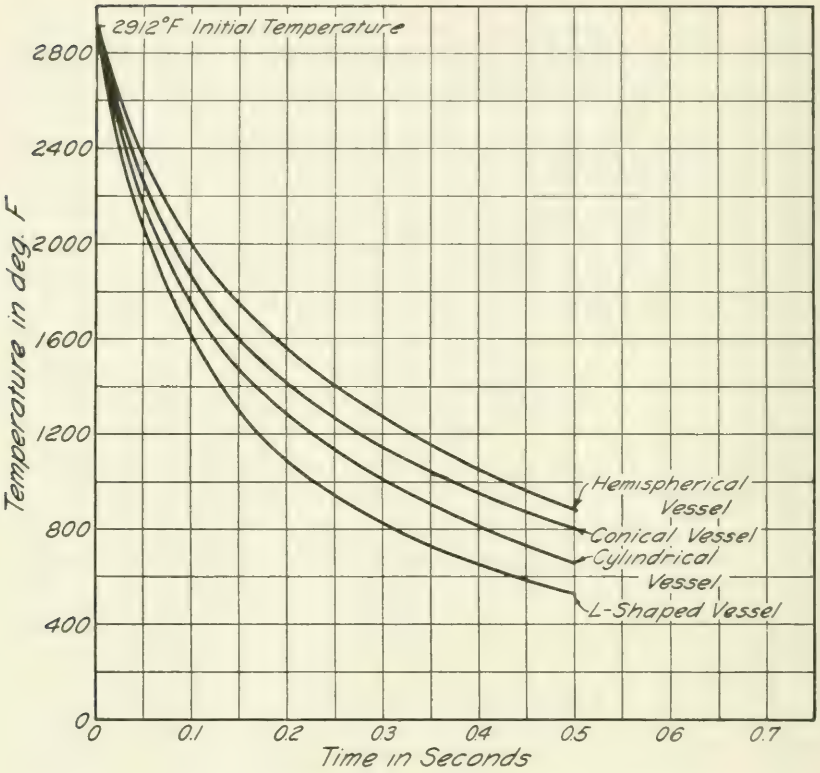


FIG. 21. RATE OF COOLING AFTER ATTAINMENT OF MAXIMUM PRESSURE

It was found that the time of explosion was greatly influenced by various conditions of turbulence and ignition, and these conditions were accordingly varied so that a wide range for the values of maximum pressures and times of explosion could be obtained. For some explosions the fan in the vessel was not run at any time. In other cases the fan was used during the admission, and in others during both the admission and the explosion. Ignition at times was accomplished by means of a single spark-gap at the vertex of the cone, and at other times by a chain of 6 spark-gaps in series extending down the axis of the vessel. These results are shown in Table 18.

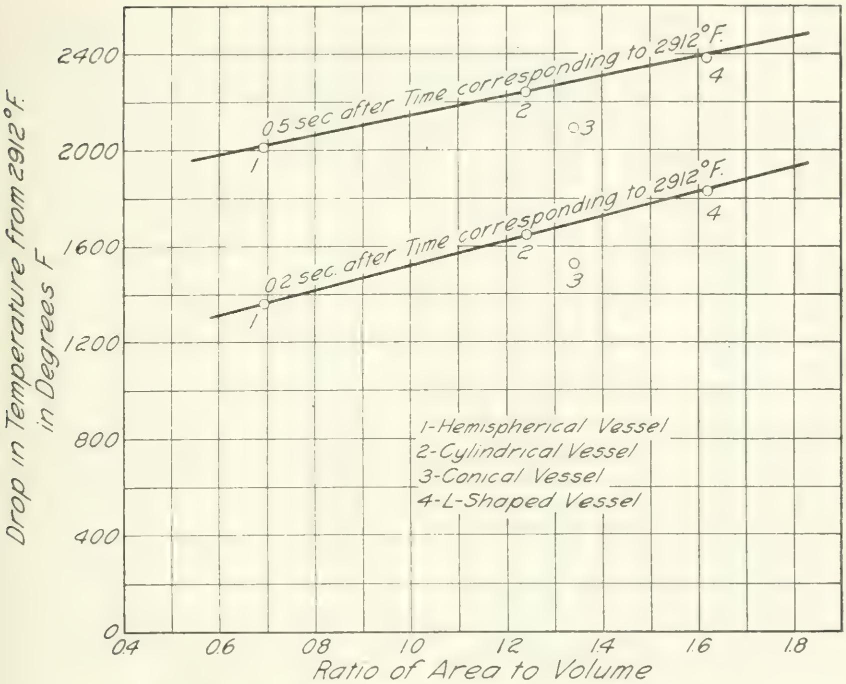


FIG. 22. CURVES SHOWING EFFECT OF AREA OF INSIDE WALLS OF VESSEL UPON RATE OF COOLING OF GAS

TABLE 4

RESULTS OF TESTS—SERIES 1

Cylindrical Vessel—Without Turbulence—Ignition at Center of Top

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
48	3.21	87.1	0.0510
49	3.77	92.5	0.0438
50	4.06	92.0	0.0366
51	4.99	84.1	0.0450
52	6.10	73.4	0.0668
53	7.03	66.2	0.0999
54	8.60	52.0	0.2390
55	9.43	44.5	0.4010
56	11.34	7.0	0.6280
57	3.62	86.5	0.0344
58	4.06	88.5	0.0370
59	4.59	87.5	0.0363

TABLE 5

RESULTS OF TESTS—SERIES 2

L-shaped Vessel—Without Turbulence—Ignition at Center of Top of Main Chamber

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
62	3.91	82.0	......
63	4.97	76.5	0.0471
64	5.79	67.5	0.0549
65	7.25	57.0	......
66	8.41	48.0	......
66a	8.13	49.1	......
67	9.71	36.5	0.3460
68	10.65	21.0	0.7220
82	2.83	....	......
83	3.26	71.0	0.1830
84	3.64	78.8	0.0578
85	4.16	82.2	0.0381
86	4.87	80.5	0.0433
87	5.80	68.0	0.0637
88	7.17	57.0	0.1218
89	8.49	47.0	0.2163
90	10.57	21.0	0.7450
134	5.87	68.1	0.0623
135	7.52	52.5	0.1455
168	8.82	40.1	0.4230
169	9.99	21.1	0.7600
170	8.09	48.0	0.2070
171	9.03	40.1	0.4180

TABLE 6

RESULTS OF TESTS—SERIES 3

L-Shaped Vessel—With Turbulence—Ignition at Center of Top of Main Chamber

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
118	3.03	80.5	0.1265
119	3.31	83.2	0.0588
120	4.20	87.0	0.0223
121	4.66	88.0	0.0187
122	4.99	84.1	0.0204
123	6.11	74.1	0.0286
124	7.01	66.0	0.0371
125	8.23	59.0	......
126	9.40	50.8	0.1057
127	10.27	48.1	0.1718
128	11.87	....	......
129	8.39	59.0	0.0604
130	4.70	88.0	0.0178
132	11.00	....	......
133	3.23	80.5	0.0702

TABLE 7

RESULTS OF TESTS—SERIES 4

L-Shaped Vessel—Without Turbulence—Ignition in Valve Chamber

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
69	3.18	61.5	0.0776
70	3.61	70.8	0.0557
71	4.06	71.0	0.0474
72	4.96	73.3	0.0504
73	5.68	66.1	0.0760
74	7.06	55.0	0.1455
75	9.26	41.5	0.3280
76	9.64	35.7	0.5100
77	10.81	12.6	0.7670
78	2.91	58.0	0.3790
79	4.14	72.5	0.0450
80	6.93	52.2	0.1179
82	4.04	74.4	0.0431
91	8.03	43.5	0.2970
92	9.12	36.5	0.4620
93	10.10	21.0	......
94	10.20	22.1	0.8070
95	11.03	6.0	0.8860

TABLE 8

RESULTS OF TESTS—SERIES 5

L-shaped Vessel—With Turbulence—Ignition in Valve Chamber

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
136	3.06	74.1	0.0455
137	2.29	78.7	0.0347
138	4.14	84.2	0.0196
139	4.76	84.2	0.0204
140	5.05	85.2	0.0209
141	6.04	74.1	0.0272
142	7.35	65.3	0.0441
143	8.32	57.0	0.0528
144	9.46	51.5	0.0728
145	10.44	49.1	0.0973

TABLE 9

RESULTS OF TESTS—SERIES 6

L-Shaped Vessel—Without Turbulence—Ignition at Center of Top of Main Chamber and in Valve Chamber

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
146	3.05	74.1	0.0538
147	3.30	75.1	0.0502
148	4.05	76.2	0.0471
149	4.75	81.5	0.0453
150	5.13	74.1	0.0501
151	6.04	65.3	0.0705
152	7.13	53.5	0.1690
153	8.50	41.0	0.2940
154	9.22	35.7	0.4495
155	10.50	17.5	0.5890

TABLE 10

RESULTS OF TESTS—SERIES 7

L-Shaped Vessel—With Turbulence—Ignition at Center of Top of Main Chamber and in Valve Chamber

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
156	3.08	78.7	0.0532
157	3.49	85.1	0.0224
158	4.18	88.0	0.0151
159	4.78	87.0	0.0175
160	4.70	87.0	0.0222
161	5.20	83.2	0.0223
162	6.21	72.5	0.0343
163	7.07	65.2	0.0390
164	8.31	57.0	0.0548
165	9.28	51.5	0.0693
166	10.40	43.6	0.1482
167	11.14	19.5	0.2660

TABLE 11

RESULTS OF TESTS—SERIES 8

Cylindrical Vessel—Without Turbulence—Ignition at Center of Top

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
206	4.17	84.5	0.0300
209	5.29	75.5	0.0930
211	6.25	64.5	0.1150
212	8.33	43.5	0.4880
214	9.37	32.5	0.8320
215	11.47	....	...
216	2.08	....	....
217	3.65	81.0	......
218	7.29	54.0	0.1720
219	2.61	67.5	0.1810
220	9.90	27.0	0.8060
221	3.65	80.0	0.0610
222	3.13	76.5	0.0580

TABLE 12

RESULTS OF TESTS—SERIES 9

Cylindrical Head—With Turbulence—Ignition at Center of Top

Test No.	Air-gas Ratio	Max. Press. lb. per sq. in.	Time of Expl. sec.
223	3.13	91.0	0.0460
225	4.17	92.0	0.0260
226	5.29	89.0	0.0330
227	6.25	78.5	0.0440
228	7.29	66.5	......
230	9.37	49.0	0.1850
231	10.42	...	......
232	11.47	..	......
233	2.08	....	......
234	2.61	83.5	0.1740
237	8.53	60.0	0.0930
238	9.90		......

TABLE 13

RESULTS OF TESTS—SERIES 10

Conical Vessel—Without Turbulence—Ignition at Vertex

Test No.	Air-gas Ratio	Maximum Press. lb. per sq. in.	Time of Expl. sec.
274	3.13	83.6	0.0710
275	2.61	80.0	0.0950
276	4.17	78.0	0.0830
277	5.21	66.5	0.1390
278	6.25	54.3	0.3290
279	7.29	40.6	0.9420
280	8.33	...	..

TABLE 14

RESULTS OF TESTS—SERIES 11

Conical Vessel—With Turbulence—Ignition at Vertex

Test No.	Air-gas Ratio	Maximum Press. lb. per sq. in.	Time of Expl. sec.
329	2.08	87.0	0.0700
330	2.61	91.5	0.0280
331	3.13	93.8	0.0390
332	3.65	90.3	0.0380
333	4.17	83.6	0.0516
334	5.21	75.5	0.0496
335	6.25	66.6	0.0810
336	7.29	56.5	0.1360
337	8.33	46.3	0.3440

TABLE 15

RESULTS OF TESTS—SERIES 12

Conical Vessel—Without Turbulence—Ignition 3 in. below Vertex

Test No.	Air-gas Ratio	Maximum Press. lb. per sq. in.	Time of Expl. sec.
306	2.61	78.0	0.0400
307	4.17	73.5	0.0790
308	5.13	81.6	0.0560
310	5.21	65.5	0.1320
311	6.25	53.5	0.2710
312	7.29	41.2	0.5500
313	8.33	....	...
314	2.08	60.0	0.1360
315	7.81	7.0	.....

TABLE 16

RESULTS OF TESTS—SERIES 13

Conical Vessel—With Turbulence—Ignition 3 in. below Vertex

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
316	2.08	80.3	0.0880
317	2.61	91.5	0.0440
318	3.13	93.8	0.0360
319	3.65	91.5	0.0380
320	4.17	87.0	0.0340
323	7.29	59.5	0.1190
324	7.81	52.0	0.1670
325	8.85	44.8	0.2670
326	9.37	5.0	......
327	5.21	79.0	0.0640
328	6.25	68.0	0.0880

TABLE 17

RESULTS OF TESTS—SERIES 14

Hemispherical Vessel—Without Turbulence—Ignition at Top

Test No.	Air-gas Ratio	Maximum Pressure lb. per sq. in.	Time of Expl. sec.
350	3.14	89.0	0.0391
351	4.20	90.5	0.0465
352	5.18	77.5	0.0732
353	6.25	66.2	0.1330
354	7.05	57.8	0.2040
355	8.22	47.2	0.4340
356	9.32	17.0	......
357	2.47	....	......
358	4.39	85.0	0.0442
359	5.76	88.6	0.0329
360	2.88	84.0	0.0625

TABLE 18

RESULTS OF EXPLOSIONS OF MIXTURE OF 1.0 PART HYDROGEN TO 2.4 PARTS AIR

Test No.	Turbulence	Ignition	Max. Pressure lb. per sq. in.	Time of Expl. sec.
1 H	During Admission	At Vertex	75.0	0.0175
2 H	During Explosion	At Vertex	78.0	0.0146
4 H	During Admission	At Vertex	86.0	0 0098
5 H	During Admission	6-gap	77.0	0.0161
25 H	During Explosion	6-gap	82.5	0.0126
26 H	None	At Vertex	84.8	0.0095
27 H	During Admission	At Vertex	79.2	0.0148

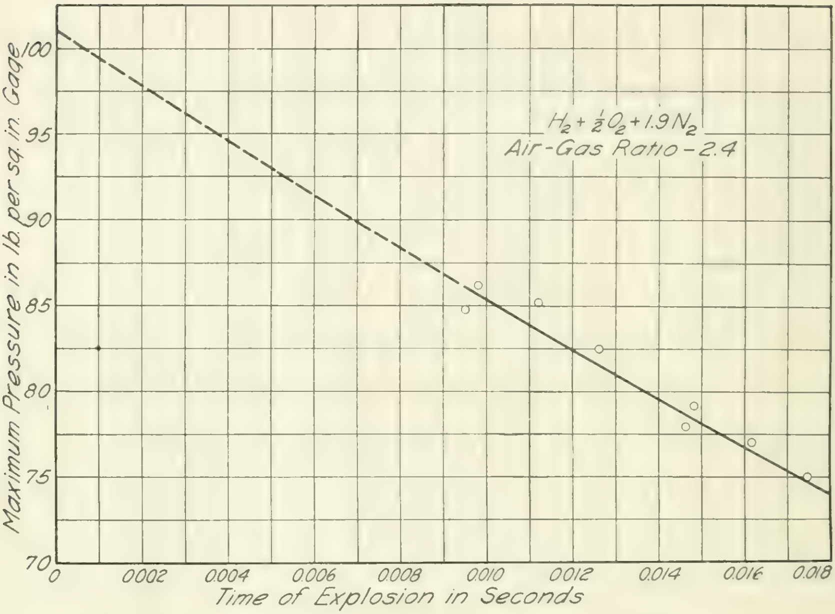


FIG. 23. MAXIMUM PRESSURE-TIME CURVE FOR EXPLOSIONS OF HYDROGEN AND AIR

Fig. 23 shows the relation between the maximum pressure and time of explosion. The relation seems to be practically linear. The variations in maximum pressures are due to variations in the heat loss to the walls of the vessel. The latter is a function of the time of explosion. If the curve is extended, as shown by the dotted portion in Fig. 23, the intercept on the zero-time ordinate represents the maximum pressure that would be produced by an adiabatic explosion of the same mixture. The maximum pressure for an adiabatic explosion has been computed in Appendix A. The computed value of 101 lb. per sq. in. gage agrees with the value of 101 lb. per sq. in. shown in Fig. 23. In Appendix A maximum pressures have also been computed for this same mixture on the assumption of heat losses ranging from 5 to 25 per cent. These results have been plotted in Fig. 24, which shows the relation between maximum pressures and heat lost

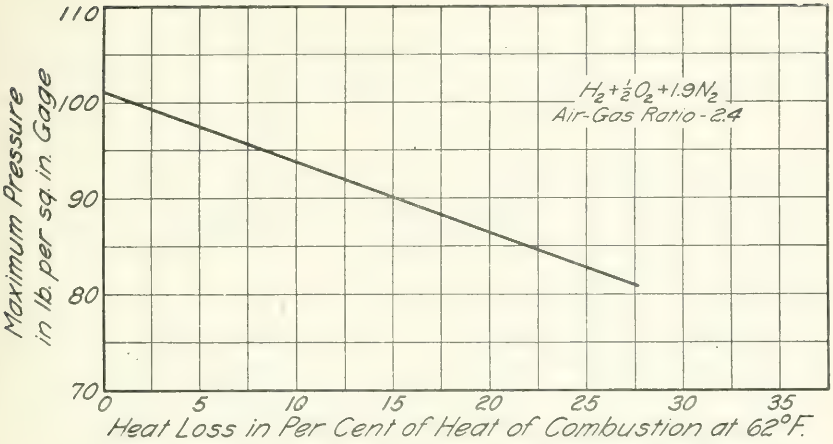


FIG. 24. THEORETICAL MAXIMUM PRESSURE AND HEAT LOSS FOR EXPLOSIONS OF HYDROGEN AND AIR

to the walls of the vessel. The highest maximum pressure observed was 86.0 lb. per sq. in. gage, and occurred on test 4H for which the time of explosion was 0.0098 sec. Reference to Fig. 24 shows a loss of 20.8 per cent of the heat of combustion of the gas charge at 62 deg. F. and 29.92 in. mercury for this maximum pressure.

The results obtained by W. T. David[211] indicate that the loss of heat by *radiation* during the explosion of a 25.4 per cent mixture of hydrogen and air is about 0.5 per cent of the heat of combustion, but he gives no estimate of the loss of heat by *conduction* to the walls of the vessel. The explosion vessel used by David, however, had a volume of about 1045 cu. in. while the one used in this investigation had a volume of 155 cu. in.; therefore the cooling effect of the walls of the vessel would be much more marked in the latter case than in the former. Moreover, the very close proximity of the walls of the conical vessel to the ignition point, which was at the vertex of the cone, would tend to cause an excessive heat loss at the beginning of the explosion. The presence of a film of water remaining on the walls of the vessel from a previous explosion would also augment the heat loss. Taking all these things into consideration, the heat loss of 20.8 per cent does not appear unreasonable for the comparatively small conical vessel.

V. CONCLUSIONS

23. *Summary of Conclusions.*—

(1) For the illuminating gas used, a ratio of air to gas of about 4 to 1 gives the maximum explosion pressure and minimum time for explosion.

(2) In general, the effect of turbulence during explosion is to cause an increase of maximum pressure and a decrease in the time of explosion. This effect is greater for lean mixtures than for the richer ones. The maximum pressure is also produced with a slightly greater air-gas ratio than for the case where no turbulence exists.

(3) The effect of turbulence seems to be due to the more intimate mixing of the gas and air before inflammation, thus bringing more molecules into contact, rather than to the projection of the flame into the unburned parts of the gas mixture.

(4) The position of the source of ignition has a considerable influence on the rate of inflammation and on the maximum pressure. In vessels patterned after the L-head type of combustion space used in internal combustion engines, ignition in the valve chamber results in a maximum pressure of about 10 per cent less than that obtained when ignition occurs at the center of the head.

(5) Both turbulence and variations in the position of ignition seem to have more influence upon the rate of inflammation than upon the maximum pressure. An explanation for this is given in Section IV, page 25.

(6) In certain cases there is some evidence of the formation of pressure waves which travel smoothly through the mixture and produce a higher maximum pressure than if the inflammation had proceeded in the usual way. These pressure waves differ in character from true explosion waves.

(7) The maximum pressure and time of explosion are materially affected by the shape of the explosion vessel. This seems to be caused by variations in the ratio of surface to volume for the different vessels. From the standpoint of maximum pressure

produced, the spherical explosion vessel is best. The cylindrical and conical vessels give about the same results, but the maximum pressure is about 8 per cent lower than that for the spherical vessel. The L-head type gives results about 16.5 per cent lower than the spherical.

(8) The combustion of the gases in any pocket in the vessel is more or less incomplete, due to the cooling effect of the walls. The incomplete combustion results in a reduction of the maximum pressure.

(9) The cooling of the gases for a given mixture during any time after the attainment of maximum pressure varies directly with the ratio of surface to volume of the explosion vessel, for the particular vessels used.

(10) Radiation from the gaseous mass is an important factor in the cooling of the mixture. Variations in the character of the inner surface of the walls of the vessel cause variations in the cooling curves.

(11) The experimental results for the maximum pressures produced by explosions of hydrogen and air agree very closely with the values computed from the properties of the gas mixture.

APPENDIX A

Calculation of Maximum Explosion Pressures of Mixtures of Hydrogen and Air

1. *Discussion of Equations.*—In order to calculate the maximum pressures produced in the explosion of any inflammable gas mixture, three factors must be taken into consideration:

(1) The heat of combustion of the mixture.

(2) The variation of the specific heats of the gases.

(3) The dissociation of the products of combustion, or rather the incomplete burning of the gases due to partially attained equilibrium at the high temperatures.

The mixture is assumed to be in a state of chemical equilibrium at the instant of maximum pressure.

The following notation will be used in the calculations:

T_1 = the temperature at equilibrium, deg. F. abs.
T_2 = initial temperature, deg. F. abs.
x = the progress of the reaction (= 1 at completion).
m = initial mixture, mols.
m' = mixture at equilibrium, mols.
U_2 = total thermal energy of the mixture at T_2, B.t.u. per mol.
U_1 = total thermal energy of the mixture at T_1, B.t.u. per mol.
u_2' = thermal energy of hydrogen at T_2, B.t.u. per mol.
u_2 = thermal energy of other diatomic gases at T_2, B.t.u. per mol.
u'_1 = thermal energy of hydrogen at T_1, B.t.u. per mol.
u_1 = thermal energy of other diatomic gases at T_1, B.t.u. per mol.
K_p = equilibrium constant.
H_v = lower heating value of the mixture at T_2, at constant volume, B.t.u. per mol.

In order to determine the two unknown variables x and T_2, two independent equations must be employed.*

---

* Credit is due to G. A. Goodenough, Professor of Thermodynamics at the University of Illinois, for the development of this method of analysis.

Equating the energies of the mixture before explosion and at the time of equilibrium (since the process is adiabatic and at constant volume),

$$U_1 + xH_v = U_2 \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

$$x = \frac{U_2 - U_1}{Hv} \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

Then for a mixture of one part of hydrogen to 2.4 parts of air

$$x = \frac{2.4u'_2 + u_2 - (2.4u'_1 + u_1)}{Hv} \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

By substituting values of T, corresponding values of x may be calculated.

The equilibrium constant K_p is a function of the temperature T and of x. It is defined in the hydrogen reaction as

$$K_p = \frac{P_{H_2O}}{P_{H_2} \cdot (P_{O_2})^{\frac{1}{2}}} \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

where P denotes the partial pressures of the various constituents as denoted by the subscripts. The following expression for K_p in terms of T has been derived from the results of a number of investigators:

$$\begin{aligned} 4.571 \log K_p = \frac{102\,100}{T} &- 6.2331 \log T_2 + 0.000236\, T_2 \\ &+ 0.0333 \cdot 10^{-6}\, T_2^2 + 1.1 \qquad . \quad . \quad . \qquad (5) \end{aligned}$$

2. *Calculations for Mixture* $H_2 + \frac{1}{2} O_2 + 1.9 N_2$.—The expression for K_p in terms of partial pressures is transformed into an expression in terms of the variable x as follows. From the reaction equation

$$H_2 + \tfrac{1}{2} O_2 + 1.9 N_2 = H_2O + 1.9 N_2$$

the initial mixture and the mixture at equilibrium are

Initial	At equilibrium
H_2....1.0 mol.	H_2O.....x mol.
O_2....0.5 mol.	H_2....$(1-x)$ mol.
N_2....1.9 mol.	O_2.....$0.5x$ mol.
	N_2.....1.9 mol.
m.....3.4 mol.	m'..$m - 0.5x$ mol.

In the equilibrium mixture the partial pressures are

$$P_{H_2O} = \frac{x}{m'} P$$

$$P_{H_2} = \frac{1-x}{m'} P$$

$$P_{O_2} = \frac{0.5x}{m'} P$$

where P denotes the total pressure of the mixture.

Substituting in equation (4)

$$K_p = \left(\frac{x}{1-x}\sqrt{\frac{m'}{0.5x}}\right)\frac{1}{\sqrt{P}} \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (6)$$

The volume remains constant; therefore the total pressure P is variable. To eliminate P combine the two equations

$$P_1 V = mRT_1 \text{ and } PV = m'RT$$

giving $\frac{P_1}{P} = \frac{mT_1}{m'T}$ or $\frac{1}{\sqrt{P}} = \frac{1}{\sqrt{P_1}}\sqrt{\frac{m}{m'}}\sqrt{\frac{T_1}{T}}$

Thus equation (6) becomes

$$K_p = \frac{x}{1-x}\sqrt{\frac{m}{0.5x}}\,\frac{1}{\sqrt{P_1}}\sqrt{\frac{T_1}{T}} \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (7)$$

Taking $T = T_2$ at equilibrium and $m = 3.4$, this equation may be written

$$\log K_p = \log\frac{x}{1-x} - \tfrac{1}{2}\log\frac{6.8}{x} - \tfrac{1}{2}\log P_1 - \tfrac{1}{2}\log\frac{T_2}{T_1} \quad (8)$$

The elimination of $\log K_p$ between equations (5) and (6) gives a second relation between x and T.

The constant 1.1 in equation (5) is based upon the atmosphere as the unit of pressure; hence, since P_1 is one atmosphere, the term $\frac{1}{2}\log P_1$ in equation (8) vanishes.

Substitution of values for T_2 in the energy equation (3) gives

T_2....	4200	4400	4600	4800	5000
x.....	0.721	0.772	0.827	0.884	0.945

Substitution of values for T_2 in the equilibrium equation gives

T_2....	4000	4400	4800	5000
x.....	0.988	0.975	0.954	0.939

These two sets of values have been plotted in Fig. 25. Reference to that figure shows that the intersection of the two curves gives values for T_2 of 4985 deg. F. abs., and for x, of 0.940, satisfying both equations.

The maximum pressure is found from the gas laws given above:

$$P_2 = \frac{m - \frac{1}{2}x}{m} \frac{T_2}{T_1} P_1 = \frac{(3.4 - 0.5 \times 0.940) \times 4985 \times 14.4}{3.4 \times 535}$$

$$= 115.5 \text{ lb. per sq. in. abs.}$$

$$= 101.1 \text{ lb. per sq. in. gage.}$$

An initial temperature of 75 deg. F. and an initial pressure of 14.4 lb. per sq. in. abs. have been assumed.

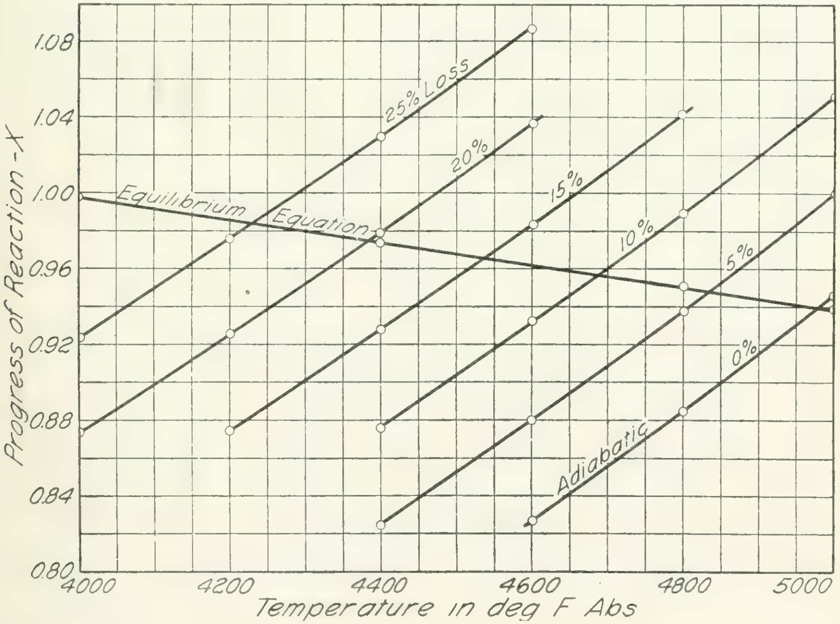


FIG. 25. GRAPHICAL SOLUTION FOR ENERGY AND EQUILIBRIUM EQUATIONS

If an amount of heat h is assumed to be lost to the walls of the vessel during explosion, the energy equation will be modified as follows:

$$U_1 + xH_v = U_2 + h$$

or
$$x = \frac{U_2 - U_1 + h}{H_v} \qquad (9)$$

The equilibrium equation is not affected by the heat loss.

The following amounts of heat loss, expressed in per cent of the heat of combustion of the mixture at 62 deg. F. (102 930 B.t.u.) have been assumed.

	B.t.u.
5 per cent	5 147
10 per cent	10 293
15 per cent	15 440
20 per cent	20 586
25 per cent	25 733

Adding these amounts into the numerator of the energy equation and solving for x and T_2 as before, several curves are obtained, crossing the curve of the equilibrium equation as in Fig. 25. From the several intersections the following results may be obtained:

TABLE 19

THEORETICAL MAXIMUM PRESSURES FOR EXPLOSIONS OF HYDROGEN AND AIR

Heat Loss Per Cent	T2 deg. F. abs.	x	P2 lb. per sq. in. gage
0	4985	0.940	101.1
5	4836	0.949	97.3
10	4688	0.957	93.9
15	4540	0.966	90.6
20	4387	0.974	86.6
25	4228	0.984	82.8

Fig. 24 (page 49) has been plotted from these results.

APPENDIX B

Historical Review of Principal Researches

1. *Researches of Dugald Clerk.*—In 1884 Dugald Clerk[31]* exploded mixtures of Cambridge coal gas and air at various initial pressures and temperatures, making use of a cylindrical explosion vessel having a volume of 317 cu. in. Explosions of hydrogen and air were also made in the same apparatus. The pressure developed by the explosion was measured by means of a Richards steam engine indicator and the pressure diagrams were traced on a revolving drum. The mixture was ignited by a spark plug placed at the bottom of the cylinder.

In a later series of tests a cylinder of 7 in. diameter and length was used. In place of the drum used on the Richards indicator a longitudinally moving slide was substituted and the recording paper was fastened to the slide. The speed of the slide was determined by means of an electrically driven tuning fork which traced a curve on the recording paper.

A set of explosion curves for different air-gas ratios, taken with Clerk's later apparatus, is shown in Fig. 26, and curves showing the

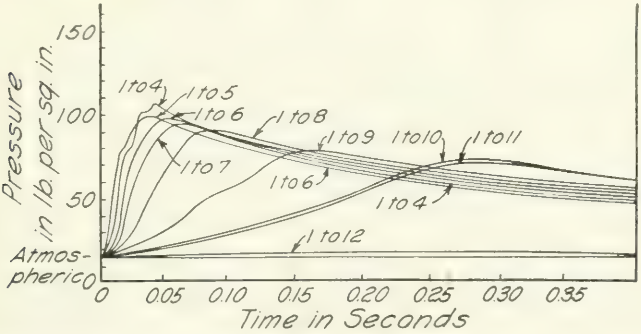


Fig. 26. Explosion Records for Coal Gas and Air—Dugald Clerk

relation between maximum pressure and time and the air-gas ratio are shown in Fig. 27. The latter are typical of the curves obtained by all the investigators in this field.

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* The small figures used in this section refer to the reference numbers given in the Bibliography in Appendix C.

Clerk's results are subject to the criticism that a piston indicator has too much inertia to record rapid changes in pressure with a high degree of accuracy. His results, however, are probably more complete and accurate than any of the other early investigators.

2. *Researches of Grover.*—In 1895 F. Grover[132] made some experiments on the explosions of mixtures of coal-gas and air. His apparatus was similar to that used by Clerk except that the explosion

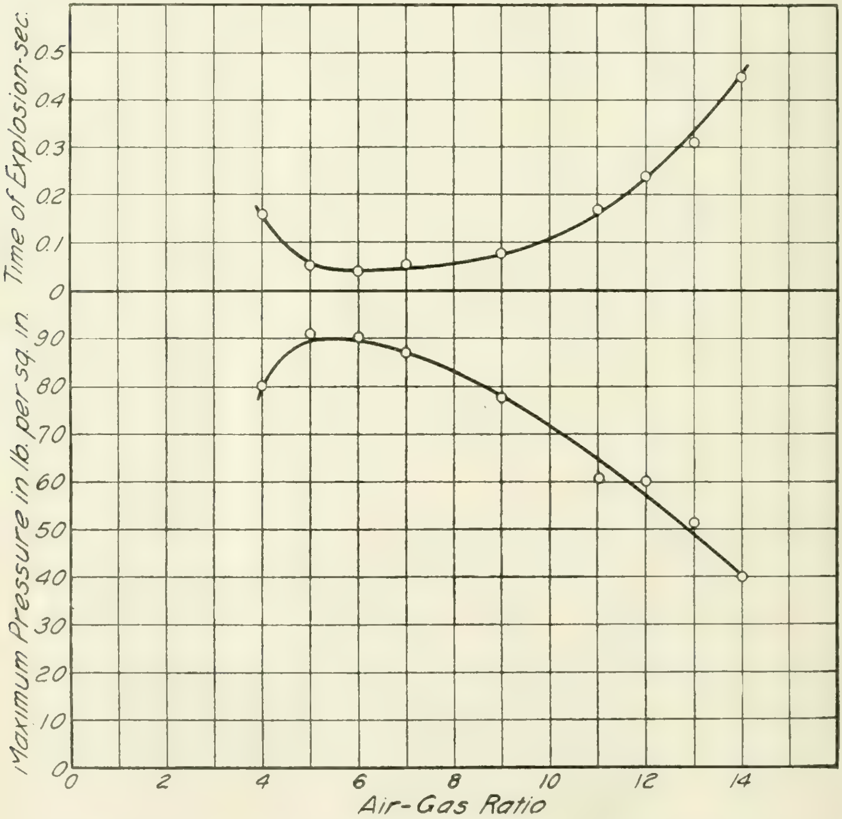

FIG. 27. EXPLOSION CURVES FOR COAL GAS AND AIR—DUGALD CLERK

vessel had a volume of 1 cu. ft. The cylinder was 8 in. in diameter and 34 in. long. The gases were introduced by filling the cylinder with water and allowing the mixture to enter as the water was withdrawn.

Grover's curves showing the relation between maximum pressure and air-gas ratio give lower values than those obtained by Clerk, and later at the Massachusetts Institute of Technology, although the calorific value of the gases used was practically the same for all cases. He attributes this difference to a reduction in the sensible heat due to the heat used in evaporating the water adhering to the cylinder walls after the gas charge had been drawn in. It is more probable, however, that the effect was due to faulty mixing of the gas charge. The time of explosion as determined by Grover was less than that found by the other experimenters. This, together with the fact that a lean mixture can be burned much more rapidly if a relatively strong mixture is situated in the immediate vicinity of the ignition point, leads to the conclusion that the lower pressures developed were caused by incomplete mixture.

Grover also studied the effect of diluting the air-gas charge with the products of combustion left from the preceding explosion. His results seemed to indicate that the presence of from 5 to 30 per cent of the exhaust gas raised the pressure as much as 19 lb. per sq. in.

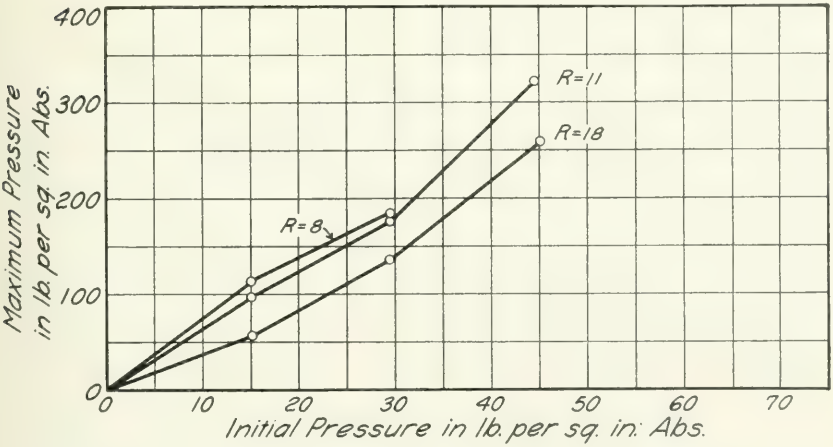

Fig. 28. Effect of Initial Pressure on Maximum Pressure for Mixtures of Acetylene and Air—Grover

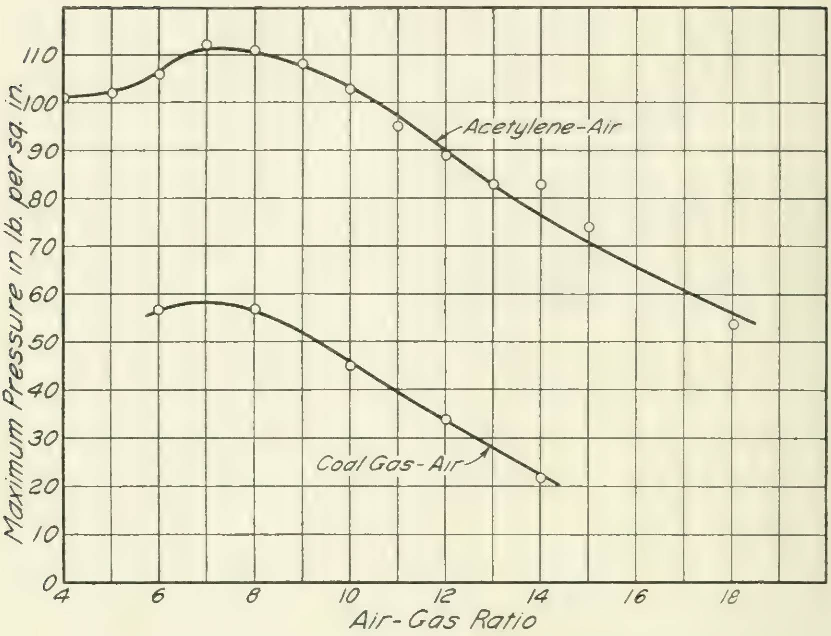

FIG. 29. EXPLOSION CURVES FOR MIXTURES OF ACETYLENE AND AIR, AND COAL GAS AND AIR—GROVER

above that observed in the explosion of a mixture containing no exhaust gas and the same percentage of fresh coal gas. This apparent increase of maximum pressure became less as the strength of the mixture became greater. A 12 to 1 mixture showed practically no increase, and for a 7 to 1 mixture the maximum pressure was decreased by the presence of the exhaust gas. An analysis of the exhaust gas used for dilution, however, proved that it contained as high as 30 per cent of combustible constituents. Since the leaner mixtures showed a greater increase in maximum pressure than the stronger ones, it is obvious that the increase was due to the presence of these added combustible gases.

In 1898 Grover used the same apparatus for a series of tests on acetylene and air mixtures. Any error due to water on the cylinder walls was eliminated by measuring the gas into the cylinder from a gas holder. Initial pressures of 1,2, and 3 atmospheres were used for mixtures of acetylene and air and for coal-gas and air. The results of the coal-gas and air tests were again low as compared with Clerk's, thus indicating the lack of a perfectly homogeneous mixture.

Curves showing the relation between explosion pressure and initial pressure are shown in Fig. 28, and others showing the relation between explosion pressure and air-gas ratio are shown in Fig. 29. In Fig. 28 the deviation from the straight-line law found by other investigators may be explained by the evidence of faulty mixing to be found in all of Grover's work.

3. *Researches at the Massachusetts Institute of Technology.*—In 1898 a series of experiments on explosions of gaseous mixtures was made at the Massachusetts Institute of Technology.[132] The apparatus used differed from Clerk's only in the recording device. A rotating disc was substituted for the slide used by Clerk, thus giving pressure curves on polar instead of rectangular coördinates. The investigation was extended to include mixtures of gasoline vapor and air. Otherwise they duplicate the results obtained by Clerk.

4. *Researches of Petavel.*—J. E. Petavel's[91] experiments in 1902 are of interest inasmuch as they are the only ones in which very high initial pressures were used. The explosion vessel was a spherical steel bomb having an internal diameter of 4 in. and a volume of 0.0195 cu. ft. The spring of the indicator consisted of a steel tube ending in a piston and loaded as a hollow column. The motion was magnified by means of a system of levers and a mirror. Since the instrument was practically free from inertia effects, it was a great improvement on the ones used by previous investigators, and was able to follow the rapid variations in pressure without appreciable lag.

5. *Researches of Bairstow and Horsely.*—In 1902 L. Bairstow and E. C. Horsely[75] experimented on different explosive mixtures, making use of the apparatus later used by Bairstow and Alexander. Their results indicate that the ratio of explosion pressure to initial pressure increases slightly with the increase in initial pressure.

6. *Researches of Bairstow and Alexander.*—In 1905 L. Bairstow and A. Alexander[90] experimented upon mixtures of coal gas and air. The gas used was city illuminating gas with an average heating value

of 628 B.t.u. per cu. ft. The explosion vessel was a cylinder 10 in. in diameter and 18 in. long. The indicator was a Simplex steam engine indicator and was, therefore, subject to the criticism that it has too much inertia.

It was found that mixtures could be allowed to stand as long as 17 hours without becoming homogeneous, and in order to overcome this difficulty they made use of a mixing plate operated by hand. Ignition was accomplished by means of a high tension spark through a firing tube. This tube which was pierced with small holes extended down into the explosion vessel. All holes were closed except the one located at the point where ignition was desired. When the spark was passed at the top of the tube, the mixture contained within the tube exploded and threw a jet of flame through the open hole, thus igniting the contents of the explosion vessel. In this way ignition at any desired point within the vessel was obtained. This method did not afford a constant intensity of ignition. When the lower holes were opened, the explosion in the tube projected the jet of flame farther into the vessel than it did when the upper holes were opened. The distance also varied with the mixture in the tube. Consequently some

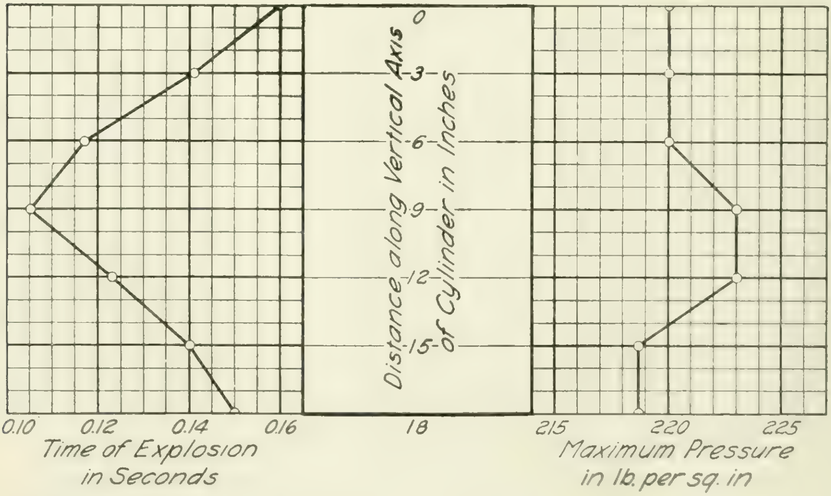

FIG. 30. EFFECT OF POSITION OF IGNITION ON MAXIMUM PRESSURE AND TIME OF EXPLOSION—BAIRSTOW AND ALEXANDER

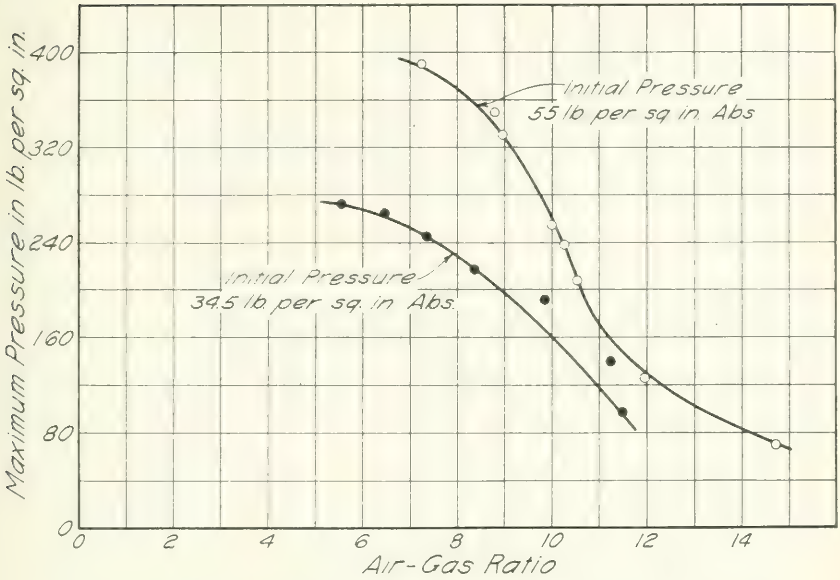

FIG. 31. EFFECT OF STRENGTH OF MIXTURE AND OF INITIAL PRESSURE ON MAXIMUM PRESSURE—BAIRSTOW AND ALEXANDER

variation in the maximum pressures and times of explosion were caused by variations in the intensity of ignition.

Fig. 30 shows the effect of the position of the ignition point upon the maximum pressure for a 7 to 1 mixture with an initial pressure of 35 lb. per sq. in. absolute. Similar tests were made with a weaker mixture, and the position of the ignition point for most rapid combustion was found to be about 3 in. below the center of the vessel. This can be attributed to the fact that the convection currents in the weaker mixture have velocity sufficient to be comparable with the rate of inflammation, and the position of the ignition point in the lower part of the vessel allows the action of the convection currents to become effective.

Two series of tests were made with varying mixtures, in all cases using four electric sparks in series passing through the axis of the explosion vessel. For one series an initial pressure of 55 lb. per sq. in. absolute, and for the other an initial pressure of 34.5 lb. per sq. in. absolute was used. The results are shown in Fig. 31. Fig. 32 clearly indicates that the maximum pressure is directly proportional to the initial pressure.

Further data on the cooling of various mixtures after explosion were given. All of Bairstow and Alexander's results, however, are subject to the criticism that variations in the intensity of ignition may have caused variations in the maximum pressure and time of explosion.

7. *Researches of Hopkinson.*—The work of B. Hopkinson[97] is distinctly significant inasmuch as he was the first investigator to make extensive use of an optical indicator for recording the explosion pressure. With this instrument he was able to record the rapid variations of pressure during explosion with a high degree of precision.

Hopkinson exploded mixtures of Cambridge coal-gas and air in a cylindrical vessel having a volume of 6.2 cu. ft. Air-gas ratios of 9 to 1 and 12 to 1 were used, and in all cases the gases were saturated with water vapor. The temperature of the burning gases was measured by means of platinum resistance thermometers inserted at different parts of the vessel as shown in Fig. 33. These thermometers consisted of short coils of platinum wire 0.001 in. in diameter. They were

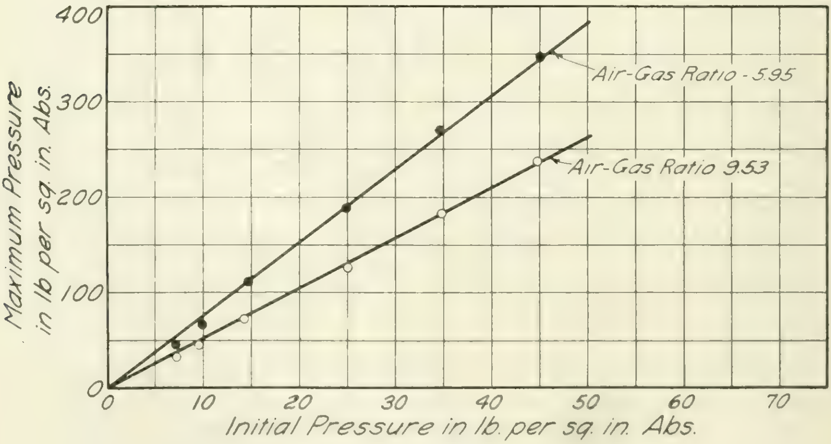

FIG. 32. EFFECT OF INITIAL PRESSURE ON MAXIMUM PRESSURE BAIRSTOW AND ALEXANDER

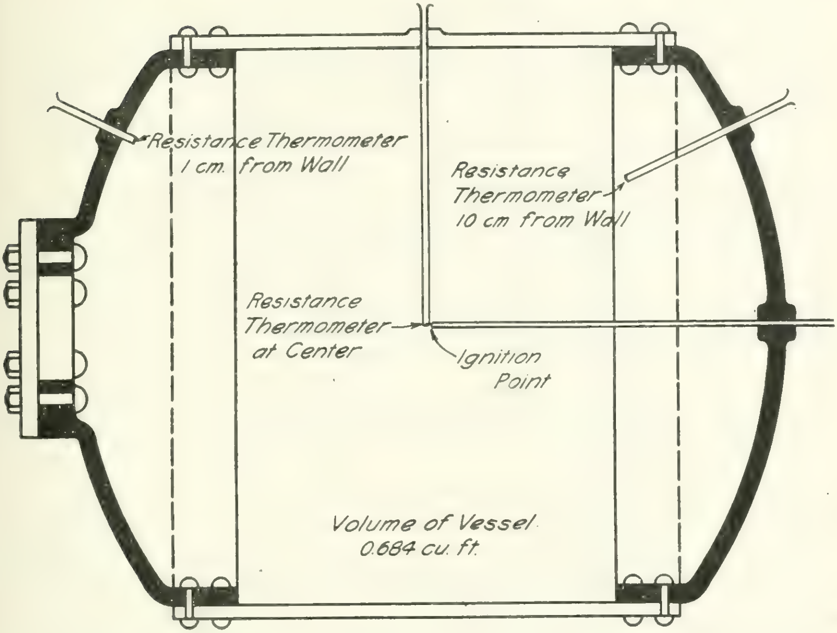

FIG. 33. EXPLOSION VESSEL—HOPKINSON

connected in series with a galvanometer, and a change in resistance of the coils, due to a change in temperature, caused a variation in the current in the circuit, thus deflecting the mirror on the galvanometer. Records of both temperatures and pressures were made by beams of light reflected from mirrors on the galvanometer and indicator to a photographic film fastened on a revolving drum. A correction was made for the time lag of the thermometers.

When the charge was fired, it was found that the coil placed at the center of the vessel almost invariably melted, thus indicating that the temperature was at least as high as 1755 deg. C. The curves in Fig. 34 showing the relation between temperature at the center of the vessel, maximum pressure, and time elapsed from ignition, also show that the center coil melted about 0.025 sec. before the attainment of maximum pressure. The temperature at the beginning of inflammation rose rapidly, indicating that the combustion of the mass of gas near the ignition point was practically complete before the pressure, as shown by the indicator, had increased appreciably.

After ignition the flame spread through the mixture with a velocity of approximately 150 centimeters per second, and completely filled the vessel about 0.03 sec. before the attainment of maximum pressure, as proved by the fact that the temperature recorded by the thermometer placed 1 cm. from the wall reached a maximum 0.03 sec. before maximum pressure. The gas at the center was then compressed adiabatically to a temperature considerably above the melting point of platinum. It is probable that this temperature was approximately 1900 deg. C. At the time of maximum pressure the temperature distribution was approximately as follows:

	Deg. C.
Mean, calculated from the pressure	1600
Center, near the ignition point	1900
10 cm. within the wall	1700
1 cm. from the wall at the end	1200
1 cm. from the wall at the side	850

It is evident, even if the explosion vessel was nonconducting and the effect of radiation and conduction was excluded, that differences in temperature would exist at different parts of the mixture due to variations in the degree of compression at the respective points.

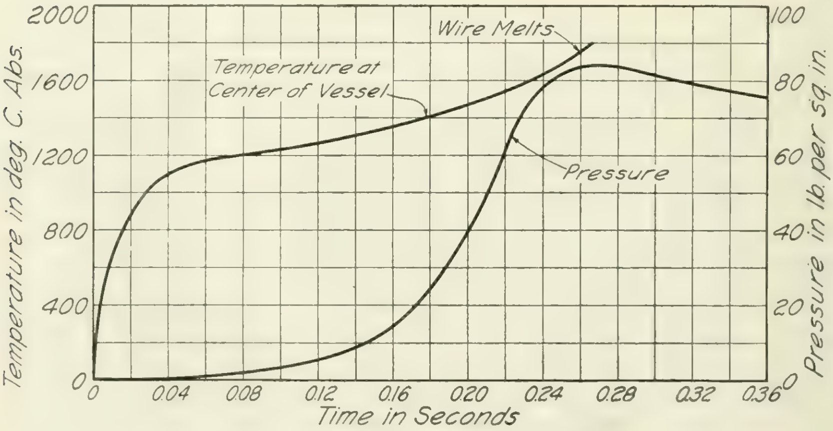

FIG. 34. EXPLOSION CURVES FOR MIXTURES OF CAMBRIDGE COAL GAS AND AIR—HOPKINSON

At a time 0.05 sec. later than the attainment of maximum pressure, the mean temperature of the gas calculated from the pressure was about 1100 deg. C. A value of 1160 deg. C. was determined experimentally by making use of a platinum resistance wire stretched entirely across the vessel.

In the explosion of 12 to 1 mixtures the effect of convection became more important because the flame was propagated more slowly. The temperature of the gas immediately below the ignition point at first increased, and then decreased, due to the upward movement of the cold unburned gases as they followed the ascending flame. At a time one second later than ignition the pressure was still less than 10 lb. per sq. in. and the upper half of the vessel was filled with burned gas which was giving up heat to the cylinder walls. The last portions of gas to be ignited were those immediately below the ignition point. The maximum pressure was 50 lb. per sq. in. and was attained 2.5 sec. after ignition. During that time at least one half of the superficial area of the vessel had been in contact with the flame. The loss of heat before the attainment of maximum pressure was, therefore, greater for a weak mixture than for a stronger one.

A later investigation[115] was undertaken with the object of measuring the heat loss to the walls of the cylinder. The explosion vessel was a cast iron cylinder one foot in diameter and one foot long, and was lined with wood. The wood was overlaid with a continuous grid of strip copper, 0.25 in. wide and 0.04 in. thick, which was connected in series with a galvanometer in order to serve as a resistance thermometer. A Hopkinson optical indicator was used to measure the pressure, and both records were photographed on a film.

The results of this series of experiments are shown in Fig. 35. Concerning these results Hopkinson says: "The heat loss begins about 0.05 seconds after ignition, when the flame first comes in contact with the copper. At first the loss goes on at a very great rate, and by the time the maximum pressure is reached, about 1700 calories, or 12 per cent of the gross heating value of the gas, has passed to the walls. The rate of heat loss at this point is about 10 calories per square centimeter per second, and the mean gas temperature is 1760 degrees Centigrade. At 0.2 seconds from ignition the rate of heat loss is about 3.5 calories per square centimeter per second, and the mean gas temperature is 1300 degrees Centigrade. This mean temperature is reduced in the ratio $\frac{74}{100}$ between these two points, but the rate of heat loss at 0.2 seconds is only one third of what it was at maximum pressure."

Hopkinson's results, obtained from these experiments, doubtless included some of the radiation loss together with the loss due to conduction. In Clerk's opinion only the conduction loss is included, but even the highly polished surface does not reflect all radiation, and the metal strip would necessarily be raised in temperatue by the radiation received.

8. *Researches of Nagel.*—In 1908 A. Nagel[128] conducted an elaborate investigation on the velocity of inflammation of various gaseous mixtures. The explosion vessel used was a spherical steel bomb 40 cm. in diameter. A diaphragm in communication with the bomb, and connected mechanically to a concave mirror, indicated the pressure developed by the explosion. Ignition was accomplished by means of

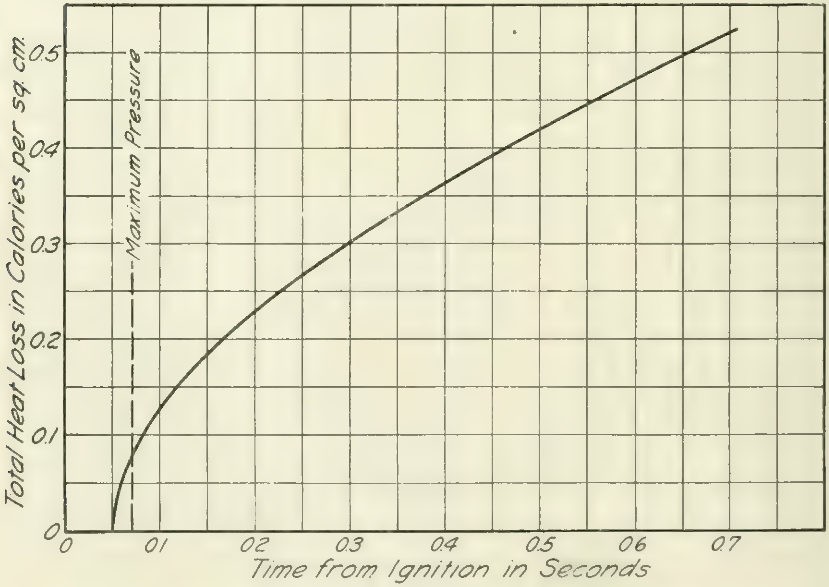

FIG. 35. HEAT LOSS IN EXPLOSIONS OF COAL GAS AND AIR—HOPKINSON

a single high tension spark at the center of the bomb. In all cases the mixture was saturated with water vapor.

Nagel's results contain no record of maximum pressures, since he was interested in the velocity of inflammation alone. For hydrogen and air mixtures he found that this velocity increased directly as the hydrogen content. With a constant mixture the velocity increased with the initial pressure, the increase being greater as the mixture became richer. For illuminating gas and for producer gas the velocity of inflammation also increased with gas content, but at constant gas content there seemed to be a tendency for the velocity to decrease with an increase in initial pressure. This effect was more marked with weak mixtures than with stronger.

The main interest in Nagel's work lies in an elaborate mathematical analysis of the rate of flame propagation in a spherical vessel. In the light of Hopkinson's work it would seem that some of Nagel's conclusions must be somewhat modified. The latter assumed that the time from the passage of the spark for ignition to the time of the attainment of maximum pressure was the time of inflammation. Hopkinson's results, however, prove that the inflammation might be complete, that is, the flame might fill the entire vessel some time previous to the attainment of maximum pressure. This would materially affect some of Nagel's conclusions relative to the rate of flame propagation.

9. *Researches of Bone.*—In 1915 W. A. Bone[172] and several of his students investigated explosions of various mixtures of hydrocarbons with oxygen and air. Two types of explosion vessels were used. One was a cylindrical vessel one foot in diameter and 8 in. long, and the other a spherical vessel 3 in. in diameter. In all cases a Petavel indicator was used to measure the pressures.

These experiments were conducted primarily to make a study of the chemical transformations involved in the explosion process, but the results were also confirmatory of the work of other experimenters on the effect of initial pressure and richness of gas mixture.

10. *Researches of David.*—Major W. T. David[144] has conducted several series of investigations on the explosions of gaseous mixtures with special reference to the cooling of the gas during explosion and the period subsequent to explosion.

In order to determine the effect of radiation, experiments were made in a cylindrical vessel 30 cm. in diameter and length. A Hopkinson indicator was used to record the explosion pressures. The radiation from the mass of gas was measured by means of a platinum bolometer placed in front of a diathermanous fluorite window located in one end. The inside walls of the vessel were either painted black, in order to absorb practically all of the heat radiated by the burning gas, or were silver plated and polished.

Tests were run in the vessel having the blackened walls, using 9 per cent and 15 per cent mixtures of Cambridge coal gas and air. Similar tests were then run in the vessel with polished walls with mixture of 13 per cent and 15 per cent gas content. From these tests the following conclusions were drawn:

(1) The total amount of heat lost to the walls of the vessel by radiation up to the time of maximum pressure is approximately proportional to the product of the third power of the maximum absolute temperature and the time of explosion.

(2) The total radiant heat lost to the walls during the explosion and the subsequent cooling is about 25 per cent of the heat of combustion of the gas.

(3) The emission of radiation at all times varies with the temperature and with the time from ignition.

(4) In weak mixtures (and probably also in stronger mixtures) the rate at which radiation is emitted is a maximum some time before the attainment of maximum pressure, and probably occurs at the time when the flame fills the vessel.

(5) Weak mixtures emit more radiation in the initial stages of cooling than strong mixtures when they have cooled to the same temperature as the weaker.

(6) Carbon dioxide emits radiation about twice as strongly as an equal volume of water vapor at the same temperature.

(7) The total heat lost by radiation up to the time of maximum pressure decreases as the initial pressure of the mixture is increased.

(8) Denser mixtures radiate heat much more strongly than those of less density, especially at the instant of maximum pressure and in the initial stages of cooling. The emission varies approximately as the square root of the density.

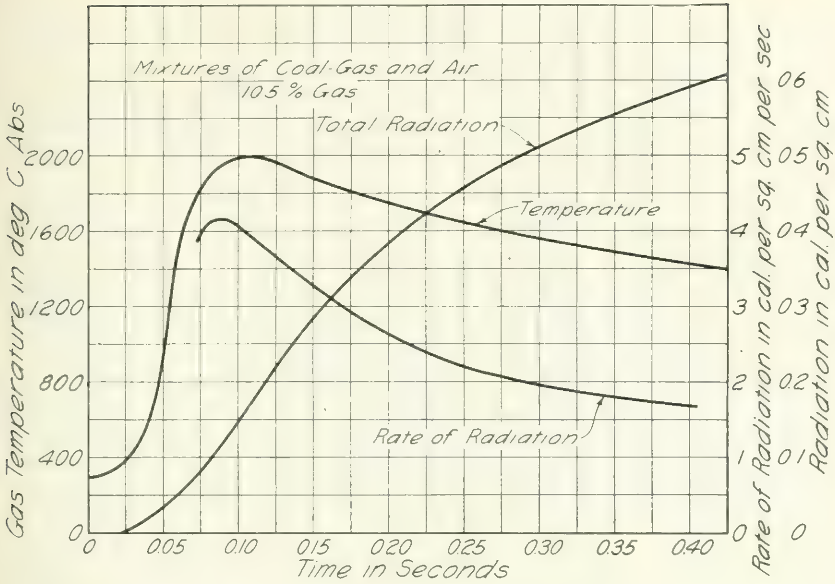

Fig. 36. Radiation Curves—David

A set of curves showing the relations between pressure, temperature, total radiation, and radiation per square centimeter of cylinder area is shown in Fig. 36, and others showing the relations between radiation loss in calories per square centimeter per second, size of vessel and initial pressure are shown in Fig. 37. The radiation loss, after correcting for heat absorbed, varies with the temperature very nearly in accordance with Planck's equation for the radiation from a flame of wave length 3.6 μ . At temperatures of 1800 to 2400 deg. C. the radiation varies with the square of the absolute temperature.

In subsequent articles David discusses the radiation from a mass of burning gas from the theoretical standpoint and analyzes his results on a purely mathematical basis.

Tests[196] were made with the same apparatus to determine the effect of diluting the mixture with carbon dioxide instead of the nitrogen in the air. Lower explosion pressures were observed when carbon dioxide was used as a diluent than when nitrogen was used. David attributes this effect to the greater specific heat of carbon dioxide at high temperature and to the fact that dissociation occurs, leaving an appreciable amount of unburned combustible in the gas at the time of maximum pressure.

David also made a study[215] of the loss of heat by conduction to the walls of the explosion vessel. A polished silver grid was mounted on a piece of linoleum and then placed on the inside of the end of the vessel. The grid was connected in series with a galvanometer in order to serve as a calorimeter. When the explosion took place the temperature of the grid increased due to the heat conducted to it from the gas. The amount of heat received by the grid in a given time could be calculated from the mass and the increase in temperature. To this was added the heat that passed through the grid into the linoleum backing, based on calculations making use of a power series formula with empirical coeffcients. The grid was located in such a manner that the heat loss as calculated corresponded to the mean heat loss by conduction for the entire vessel.

The loss of heat by conduction up to the time of maximum pressure

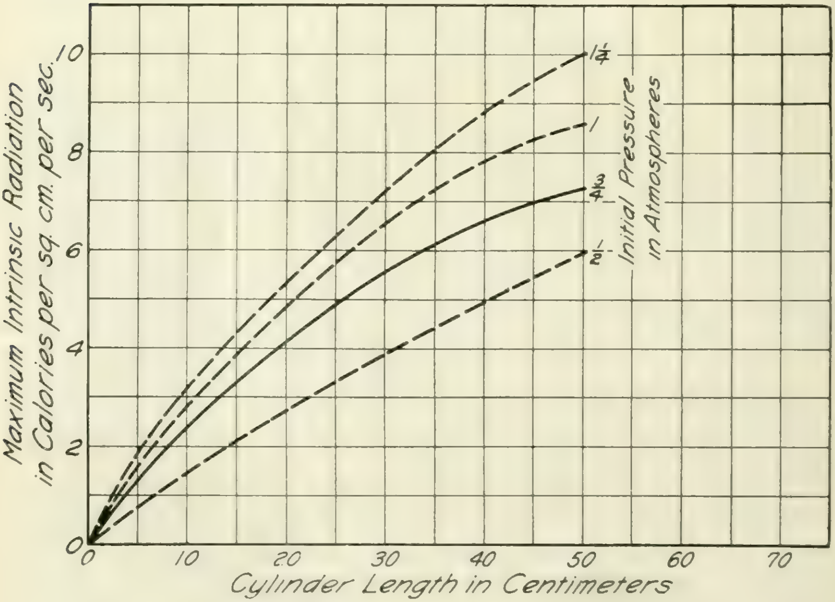

Fig. 37. Effect of Initial Pressure and Cylinder Dimensions on Radiation—David

TABLE 20

HEAT LOSS BY CONDUCTION—DAVID

Per Cent Gas in Mixture	Per Cent Heat of Combustion Lost
15.0	5.1
12.4	5.5
9.7	11.0

expressed as a per cent of the heat of combustion of the gas mixtures used is given in Table 20.

The greater time required for the explosion of the weaker mixtures seems to influence the total radiation more than the higher temperatures attained in the case of the stronger, since an increase in heat loss is observed as the mixture becomes leaner.

At 0.5 sec. after ignition the 15 per cent mixture showed a loss by conduction of 38 per cent of its heat of combustion. At the same time the 12.4 per cent mixture had lost about 34 per cent and the 9.7 per cent mixture about 28 per cent of their heats of combustion. Fig. 38 shows the rate of heat loss by conduction during the cooling

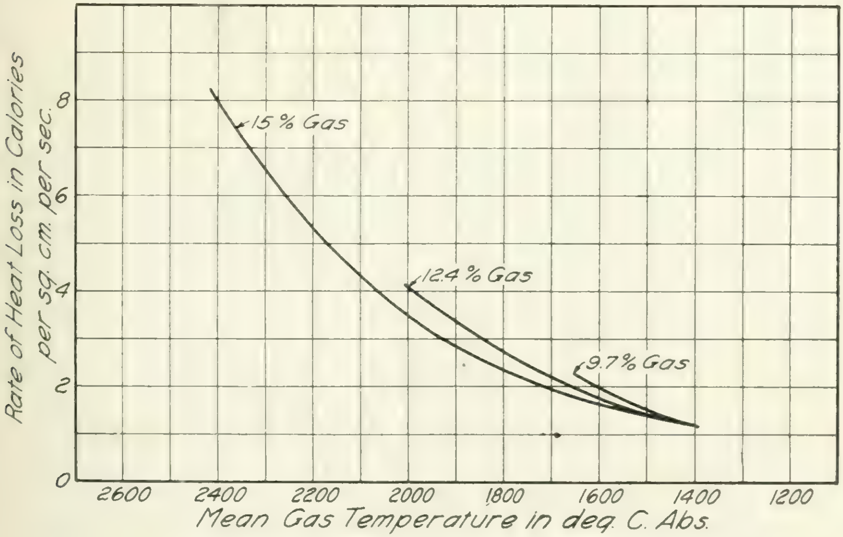

FIG. 38. HEAT LOSS BY CONDUCTION—DAVID

process. In the initial stages of cooling the weaker mixtures seem to lose heat more rapidly than the stronger mixtures when they have cooled to the same temperature. This is probably due to the action of convection currents, which are more effective in the early stages of cooling than in the later.

No mention is made in David's report of any correction having been applied for radiant heat absorbed by the silver grid. Highly polished silver absorbs about 4 per cent of the radiant heat falling upon it,* and it is probable that the grid actually in use absorbed more than 4 per cent due to unavoidable deposits on the surface.

In a later paper David[216] combined the losses due to radiation and conduction, and formed an estimate of the internal energy of the mixture during combustion. The data of this paper indicate that at the moment of maximum pressure about 90 per cent of the heat of combustion of the mixture has been converted into thermal energy, and that after-burning continues for at least 0.025 sec. after maximum pressure is reached. The distribution of energy at the moment of maximum pressure is as follows:

Internal thermal energy — 72 to 80 per cent.
Available chemical energy — 10 per cent.
Heat lost to walls — 10 to 18 per cent.

David has also measured the radiation emitted in explosions of hydrogen and air.[211]

11. *Miscellaneous Researches.*—A number of researches on gaseous explosions have been conducted by other experimenters but those discussed are characteristic of all the work in the field and have been chosen as the most representative.

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* Physikalische-Chemische Tabellen, by Landolt and Börnstein, p. 961.

APPENDIX C

Bibliography on Gaseous Explosions

1. *General Statement.*—This bibliography contains references, not only on the subject of the physical phenomena involved in the explosions of gaseous mixtures as considered in this report, but also in regard to the chemical transformations and other allied phenomena. Numerous references of a mathematical nature, particularly in regard to the subject of explosion waves, are also included, as well as some few references describing instruments and apparatus used in investigations of gaseous explosions.

2. *Abbreviations of Titles of Periodicals.*—The titles of various periodicals have been abbreviated when quoted in the Bibliography; therefore the following reference table of abbreviations and full titles is given:

Ann. Chim. et Phys.—Annales de Chimie et Physique.
Ann. des Mines—Annales des Mines.
Ann. der Phys.—Annalen der Physik und Chemie.
Ber. deutsch Chem. Gess.—Berichte der deutschen Chemische Gessellschaft.
Brit. Assn. Adv. Sc.—British Association for the Advancement of Science.
Bul. Soc. l'Ind. Minerale.—Bulletin—Société de l'Industrie Minérale.
Chem. News.—Chemical News and Journal of Physical Science.
Compt. Rend.—Comptes Rendus de l'Académie des Sciences.
Engr.—The Engineer (London).
Eng.—Engineering (London).
Eng. News.—Engineering News.
Jour. Am. Chem. Soc.—Journal of the American Chemical Society.
Jour. A.S.M.E.—Journal of the American Society of Mechanical Engineers.
Jour. Chem. Soc.—Journal of the Chemical Society (London).
Jour. für Gasbel.—Journal für Gasbeleuchtung.
Jour. Prakt. Chem.—Journal für Praktische Chemie.
Jour. S. A. E.—Journal of the Society of Automotive Engineers.
Mem. Manchester Lit. and Phil. Soc.—Memoirs, Manchester Literary and Philosophical Society.
Mitt. ü. Forsch.—Mitteilungen über Forschungsarbeiten.
Phil. Mag.—London, Edinburgh and Dublin Philosophical Magazine and Journal of Science.
Phil. Trans. Roy. Soc.—Philosophical Transactions of the Royal Society.

Phys. Zeit.—Physikalische Zeitschrift.

Proc. Inst. C. E.—Proceedings, Institution of Civil Engineers.

Proc. Inst. M. E.—Proceedings, Institution of Mechanical Engineers.

Proc. London Phys. Soc.—Proceedings of the London Physical Society.

Proc. Roy. Soc.—Proceedings of the Royal Society.

Revue Gen. Elect.—Revue Générale de Électricité.

Sitz. Akad. Wiss. Wein—Sitzungsberichte der Akademie Wissenschaften zu Wein.

Tech. Paper, U. S. Bureau of Mines—Technical Paper, United States Bureau of Mines.

Zeit. Elektro.—Zeitschrift für Elektrochemie.

Zeit. Phys. Chem.—Zeitschrift für Physikalische Chemie.

Zeit. ver. d. Ing.—Zeitschrift des vereines deutscher Ingenieurs.

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UNIVERSITY OF ILLINOIS
ENGINEERING EXPERIMENT STATION

BULLETIN NO. 134 NOVEMBER, 1922

AN INVESTIGATION OF THE PROPERTIES OF CHILLED IRON CAR WHEELS

PART II

WHEEL FIT, STATIC LOAD, AND FLANGE PRESSURE STRAINS
ULTIMATE STRENGTH OF FLANGE

CONDUCTED BY
THE ENGINEERING EXPERIMENT STATION
UNIVERSITY OF ILLINOIS

IN COÖPERATION WITH
THE ASSOCIATION OF MANUFACTURERS OF CHILLED IRON CAR WHEELS

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ENGINEERING EXPERIMENT STATION
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AN INVESTIGATION OF THE PROPERTIES OF CHILLED IRON CAR WHEELS

PART II. WHEEL FIT, STATIC LOAD, AND FLANGE PRESSURE STRAINS—ULTIMATE STRENGTH OF FLANGE

I. Introduction

1. *Preliminary Statement.*—This bulletin presents two additional phases of the investigation of the properties of chilled iron car wheels conducted at the University of Illinois. This investigation had for its object the determination of the strains which may occur within a car wheel and the limitations of present designs, with the view of improving the chilled iron car wheel and making it more satisfactory under present and future service requirements. The work was done under a coöperative agreement between the Association of Manufacturers of Chilled Car Wheels and the Engineering Experiment Station of the University of Illinois.

2. *Scope of Tests.*—A previous bulletin* dealt with the strains in chilled iron car wheels caused by forcing the wheel upon its axle and then applying a static load.

The first part of the present bulletin contains a report of an extension of the tests previously reported in Bulletin No. 129 and discusses the strains existing in a 33-inch 840-pound Arch Plate Wheel† when subjected to the combined effect of mounting, static load, and side thrust pressures; the remainder of the bulletin deals with a series of tests made to determine the ultimate strength of the car wheel flange. This latter series of tests is concerned with 33 wheels having flanges of various shapes and includes specimens representative of both new and worn wheels. Among them will be found wheels with the standard contour recommended by the Master Car Builders' Association, and others with the flange suggested by the Association of Manufacturers of Chilled Car Wheels to replace the present M. C. B. standard flange.* In addition there are wheels with flanges ground to represent various stages of wear and also two wheels which were worn out in service.

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* "An Investigation of the Properties of Chilled Iron Car Wheels. Part I: Wheel Fit and Static Load Strains," by J. M. Snodgrass and F. H. Guldner. Univ. of Ill. Eng. Exp. Sta. Bul. 129, 1922.

† For description of Arch Plate Wheel see, "An Investigation of the Properties of Chilled Iron Car Wheels. Part I: Wheel Fit and Static Load Strains," by J. M. Snodgrass and F. H. Guldner. Univ. of Ill. Eng. Exp. Sta. Bul. 129, Appendix A, pp. 75-77, 1922.

3. *Acknowledgments.*—The agreement, under which the investigation of the properties of chilled iron car wheels, of which this bulletin constitutes a partial report, was carried on, essentially provided that the Association of Manufacturers of Chilled Car Wheels furnish funds to defray the expenses incident to carrying on the investigation, while the Engineering Experiment Station of the University of Illinois conduct the research, provide the use of available laboratories, shops and office facilities, and publish a report of the results. Bulletin No. 129† presents the form of the agreement under which the work was carried out.

The Association of Manufacturers of Chilled Car Wheels appointed GEORGE W. LYNDON and F. K. VIAL, President and Consulting Engineer, respectively, of the Association, as a committee to assist in promoting the investigation. In addition to the financial help given by the Association, the investigation owes much to the assistance rendered by this committee in the way of technical advice, helpful interest, and effective coöperation throughout the progress of the work. The Engineering Experiment Station of the University of Illinois appointed EDWARD C. SCHMIDT, Professor of Railway Engineering, and ARTHUR N. TALBOT, Professor of Municipal and Sanitary Engineering, as a committee in charge of the investigation. Professor Schmidt withdrew from the University to enter the United States Military Service during the late war, and was replaced on this committee by JOHN M. SNODGRASS, Professor of Railway Mechanical

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* Some years ago the Association of Manufacturers of Chilled Car Wheels suggested that the Master Car Builders' Association increase the thickness of the flange by adding 3/16 inches of metal (about 15 lb.) to the inner side of the flange so as to increase its strength in order to care for increasing wheel loads. To date the Master Car Builders' Association has not admitted the desirability of or the necessity for increasing the thickness of the flange.

† "An Investigation of the Properties of Chilled Iron Car Wheels, Part I: Wheel Fit and Static Load Strains," by J. M. Snodgrass and F. H. Guldner. Univ. of Ill. Eng. Exp. Sta. Bul. 129, 1922.

Engineering. In the direction of the investigation this committee has had frequent conferences with the committee of the Association of Manufacturers of Chilled Car Wheels.

The greater part of the experimental work was carried on by F. H. GULDNER, who became connected with the work a few months after its inception. R. E. TURLEY and O. S. BEYER, JR., both of whom were connected with the experimental work during the earlier part of the investigation, did much in the way of developing methods, making some early tests, and getting the work under way. The committees are further indebted to H. F. MOORE, Professor of Engineering Materials at the University of Illinois, for his assistance and advice upon various matters relating to the investigation.

II. General Statement of Flange Problem

4. *Causes and Magnitude of Flange Pressure.*—When a train is traveling on a tangent track the flanges of the wheels, due to lateral oscillation of the train, strike against the rails alternately on one side and the other; when the train is traveling on a curved track, the wheel flanges are forced against either the inner or outer rail depending on the speed and the amount of superelevation of the outer rail. In both cases the car wheel flanges in contact with the rail are factors in transmitting the various forces set up within the car to the rail. In general, since the flange pressures produced in traversing curves are the more severe, the pressures arising on tangent track need not be considered. The flange by functioning in this manner is the primary cause of the following effects in the wheel itself: First, the pressure or force between rail and flange sets up internal strains and stresses within the body of the wheel; in the case of a wheel subjected to severe operating conditions the internal stresses may result in the failure of the wheel. Second, the pressure between rail and flange coupled with the rotation and sliding of the wheel on the rail results in a grinding or wearing action and in the consequent reduction in area and strength of the flange. Third, the rail pressure, when of sufficient magnitude, and if acting on a flange materially worn, may cause the failure of the flange.

The problem of calculating the force between rail and flange under various conditions is an intricate one and cannot be readily solved as a problem in mechanics on account of the many variables involved. An approximate method of estimating the flange pressure, however, has been suggested by A. M. Wellington* and F. K. Vial,† who state, briefly, that maximum flange pressure may equal one and one-half times the static load on the wheel. The use of this method probably gives higher values of flange pressure than would be attained in service and thus errs on the side of safety. Experimental methods have also been tried to determine flange pressures. An examination of the experimental methods used indicates that although flange pressures were measured, no evidence exists that they were the maximum pressures that would be encountered under the test conditions. Until more definite methods are devised it seems that the Wellington-Vial method of estimating flange pressure is the most satisfactory at present available.

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* Wellington, A. M. "The Economic Theory of Railway Location." 6th Edition, page 281, 1911.

† "A General Survey of the Mechanics of the Chilled Iron Wheel." Geo. W. Lyndon and F. K. Vial, Proc. Pittsburgh Railway Club, page 25, May 25, 1917.

As no satisfactory analysis has as yet been made of the strength of the wheel flange or the ability of the plates of the wheel to withstand internal stress due to flange pressure, recourse must be had to experimental methods to obtain information with respect to these particulars.

III. APPARATUS AND METHODS USED IN MAKING TESTS OF COMBINED EFFECT OF MOUNTING, STATIC LOAD, AND FLANGE PRESSURE

5. *Apparatus Used in Determining Strains.*—The equipment used to apply the loads for experimentally determining the combined effect of wheel fit, static load and flange pressure is shown in Fig. 1. It consisted of a 200-ton hydraulic jack supported on rolls which in turn were supported on the bed of a 600 000-lb. testing machine. Specially made castings fastened to each end of the jack had hardened tool steel blocks inserted near their ends, the upper sides of these blocks having contours similar to the head of a rail. A pair of wheels previously mounted on an axle rested on these steel blocks. By means

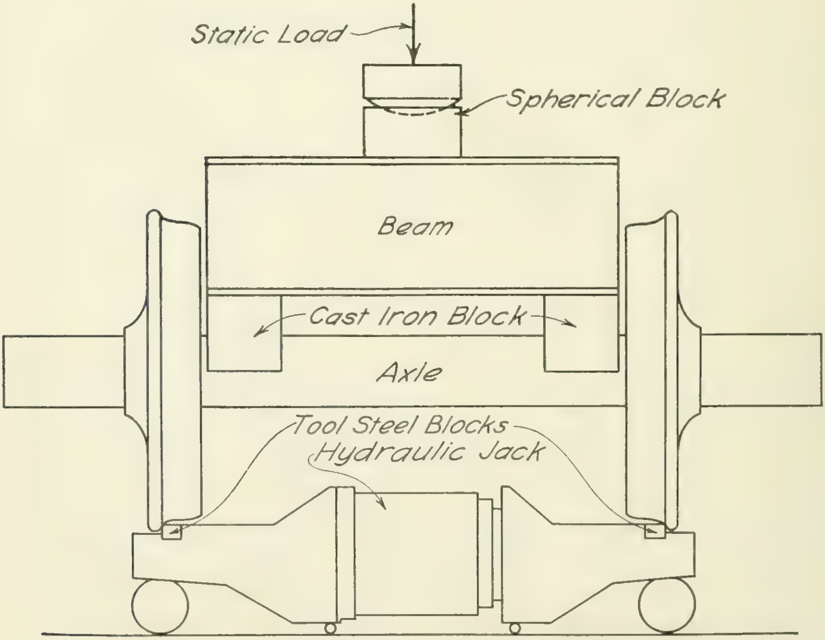

FIG. 1. METHOD OF APPLYING STATIC LOAD AND FLANGE PRESSURE

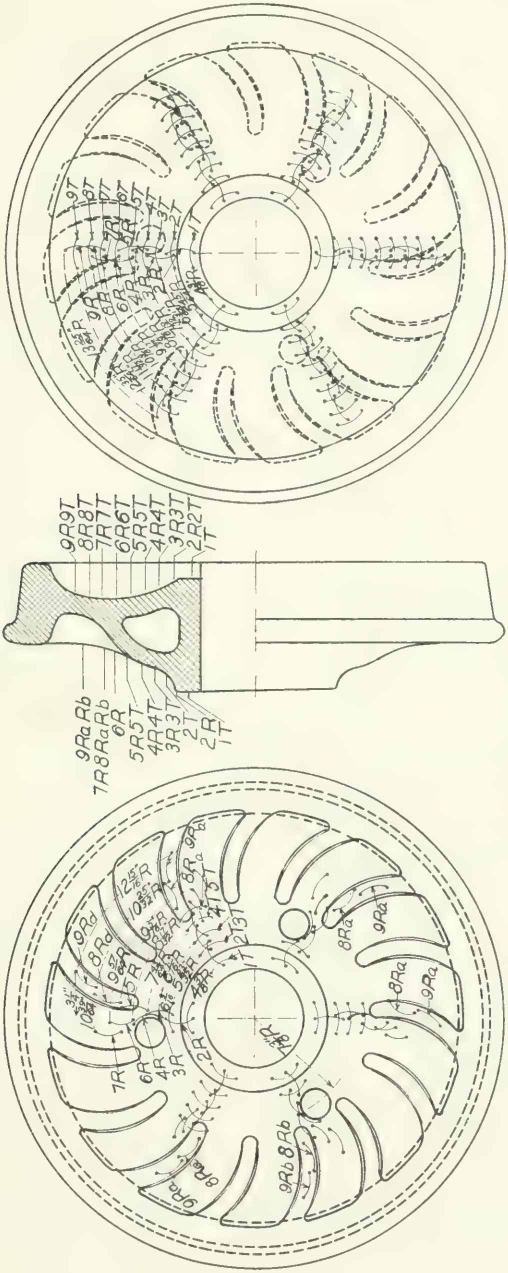

FIG. 2. LOCATION OF GAGE-LINES ON 33-IN. 840-LB. ARCH PLATE WHEEL No. 942 805

of a block with a spherical seat, a beam, and two cast iron blocks resting on the axle, as shown in the figure, the static load was transmitted to the axle, thence through the wheels and on down to the bed of the testing machine. After a static load of the desired magnitude had been applied, a flange pressure of the desired amount was obtained by means of the hydraulic jack. The strains due to the various combinations of load were determined by means of a Berry strain-gage.

6. *Preparation and Method of Testing.*—One 33-in. 840-lb. Arch Plate wheel was tested for the combined effect of wheel fit, static load, and flange pressure. The wheel was prepared with 2-in. gage-lines as shown in Fig. 2. The gage-lines were numbered from hub to tread; the numerals were suffixed with either R or T, signifying that the gage-line was in a radial or tangential direction respectively. The initial or before-load readings were taken with the strain-gage. The wheel was then pressed on the axle and a second set of strain-gage readings obtained. As previously explained* the difference between the initial readings and those taken after mounting are the strains produced by mounting the wheel on its axle. After the strains caused by the wheel fit had been thus determined, a second wheel was pressed on the axle to serve as a dummy in the application of additional loads. The two wheels, the hydraulic jack, and the remainder of the apparatus were then assembled in the testing machine as shown in Fig. 1. A net static load of 13 050 lb. was applied to the 840-lb. Arch Plate wheel and a third series of strain readings taken. The differences between these readings and the initial readings represent the strains due to the combined effect of wheel fit and a 13 050-lb. static load. In a similar way the effect of wheel fit and a 25 450-lb. static load was determined. A side thrust of 8000 lb. was then added by means of the hydraulic jack and a fresh set of strain-gage readings obtained. The latter measurements represent the combined effect of mounting, 25 450-lb. static load, and 8000-lb. flange pressure. Keeping the static load equal to 25 450 lb., additional readings were taken with side thrusts of 16 000 lb. and 32 000 lb. respectively. It seems reasonable to suppose that the latter conditions of load do not differ materially from the normal maximum loading that might occur in service, and

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* "An Investigation of the Properties of Chilled Iron Car Wheels, Part I: Wheel Fit and Static Load Strains," by J. M. Snodgrass and F. H. Guldner. Univ. of Ill. Eng. Exp. Sta. Bul. No. 129, p. 33, 1922.

the strains in the wheel would therefore approximate those that would be present under the combination of normal wheel fit, maximum static load, and maximum flange pressure.

7. *Evaluation of Stresses Existing in a Loaded Car Wheel.*[*] —It is evident that the problem of the car wheel under load is one of compound stress; that is, internal stress or force in more than one direction. In the tests herein discussed the fundamental data are measured deformations or strains due to the effect of more than one stress acting within the material; but throughout this report the results are presented and discussed as computed stresses, the term stress denoting the internal force that would exist in the material if subjected to

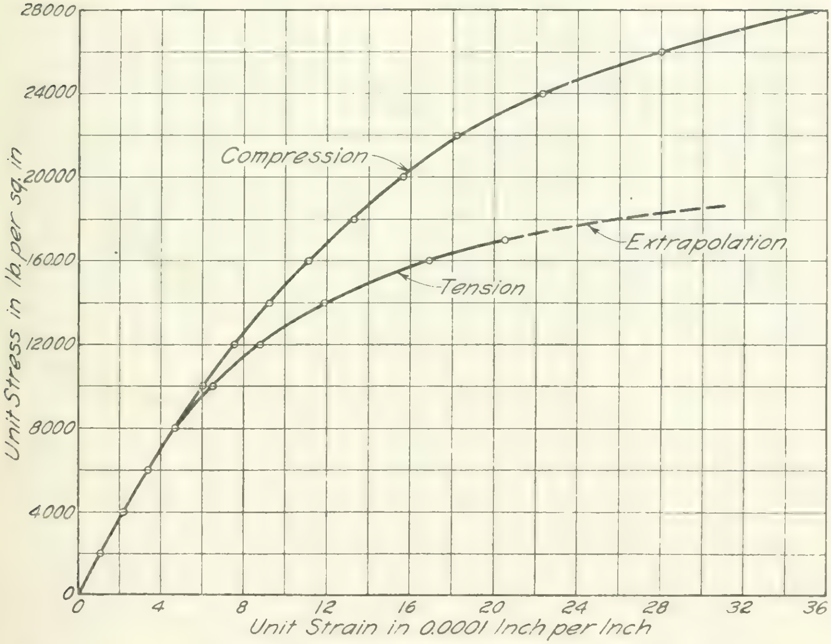

FIG. 3. ASSUMED MEAN STRESS-STRAIN RELATION OF CHILLED CAR WHEEL IRON

either simple tension or simple compression, and, under these conditions, deformed to an extent equal to the measured strain. In order to call attention to this, the computed stresses are often expressed as "corresponding simple stresses," or in other words this term and stress have identical meanings herein.

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[*] See also "An Investigation of the Properties of Chilled Iron Car Wheels, Part I: Wheel Fit and Static Load Strains," by J. M. Snodgrass and F. H. Guldner. Univ. of Ill. Eng. Exp. Sta. Bul. No. 129, pp. 15-29, 1922.

The corresponding simple stress is not the real stress or force that would be used in the calculation of the equilibrium of the forces. Its use, however, will give the reader a readier method of comparison than would the use of the measured strains. The computed stresses as presented and discussed were determined from the measured strains and average stress-strain curves plotted from tests in simple tension and compression of a number of specimens taken from a chilled iron wheel. As no two specimens even when cut from the same wheel give identical stress-strain relations, and as it would obviously be impracticable to cut a test specimen from each position in the wheel at which strains were measured, it was necessary to obtain average curves which could be considered as fairly representative of chilled car wheel irons in tension and compression, respectively. For this purpose, 13 specimens were tested in simple tension, and 12 in simple compression. The data thus obtained were averaged, and from the means the tension and compression curves shown in Fig. 3 were drawn. These curves are assumed to be representative tensile and compressive stress-strain curves for chilled wheel iron and are the curves upon which the stresses hereinafter given are based.

IV. Results of Tests

8. *Strains and Corresponding Simple Stresses Due to Mounting Wheel on Axle.*—The pressure required to mount the wheel on the axle together with the fit allowance is given in Fig. 4.

For wheel seat diameters of 6.5 and 7.0 inches the Master Car Builders' Association recommends final mounting pressures ranging from 40 to 60 and 45 to 65 tons, respectively. For a 6.75 inch bore, by interpolation, the recommended pressure range would be from 42.5 to 62.5 tons. To keep within these limits the manufacturers ordinarily make the diameter of the axle at the wheel seat 0.0105 to 0.0165 inches larger than the bore of the wheel. In this particular wheel the wheel seat diameter was roughly equal to 6.75 inches, and the allowance was 0.019 or 0.0025 inches greater than the manufacturers' maximum, while the final mounting pressure was 43.4 tons, or about 25 tons less than might be expected from this fit allowance. A possible explanation of this difference is that the mounting pressure may be a function of the speed of mounting.

The measured strains due to mounting the wheel on the axle together with those caused by mounting in combination with static load and flange pressure are given in Table 1, while the computed stresses corresponding thereto are plotted in Fig. 5.

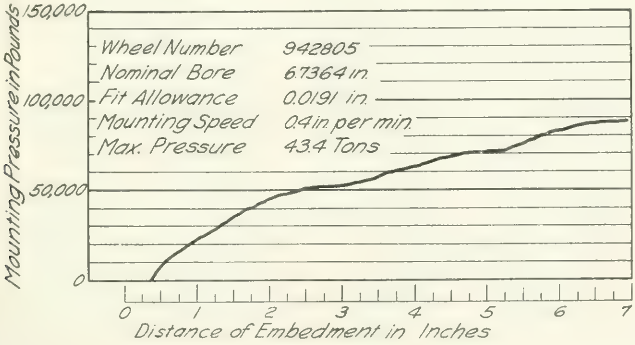


Fig. 4. Autographic Diagram of Pressure Required to Mount 33-in. 840-lb. Arch Plate Wheel No. 942 805

TABLE 1

UNIT STRAINS PRODUCED BY COMBINED EFFECT OF MOUNTING, STATIC LOAD AND FLANGE PRESSURE

33-in. 840-lb. Arch Plate Wheel No. 942 805

Strains Given in Table Are Expressed in Hundred-thousandths of Inches Per Inch

The minus sign denotes a compressive strain; an absence of sign indicates a tensile strain

LOAD APPLIED AT A

On the Outer Face of the Wheel

Gage-Line	Effect Due to Mounting Alone	Combined Effect Due to Mounting, Static Load and Flange Pressure				
		Static Load, in Pounds				
		13 050	25 450	25 450	25 450	25 450
		Flange Pressure, in Pounds				
		0	0	8 000	16 000	32 000
2R	—33	—28	—34	—34	—39	— 5
3R	—42	—27	—39	—42	—29	— 5
4R	—72	—56	—73	—44	10	33
5R	—54	—56	—54	—39	43	86
6R	....	8	—10	—10	39	105
7R	— 7	8	—29	— 5	—12	38
8R	5	17	10	— 5	0.0	57
9R	— 2	....	—12	....	—27	36
1T	66	77	75	60	69	74
2T	61	56	....	72	71	71
3T	36	56	33	32	....	27
4T	40	40	50	69	113	59
5T	11	80	43	37	74	39
6T	3	66	28	8	26	13
7T	9	66	14	—16	— 2	0.0
8T	21	38	18	22	32	21
9T	....	10	0.0	—35	....	—41

On the Inner Face of the Wheel

Gage-Line	Effect Due to Mounting Alone	Combined Effect Due to Mounting, Static Load and Flange Pressure				
		Static Load, in Pounds				
		13 050	25 450	25 450	25 450	25 450
		Flange Pressure, in Pounds				
		0	0	8 000	16 000	32 000
2R	—58	—29	—15	—39	—97	—116
3R	—35	—48	—109	—111	—53	—92
4R	20	—10	— 5	—14	—14	—53
5R	40	40	83	—23	—30	—35
6R	23	7	— 3	....	—13	—26
7R	—16	—14	....	—28	—38	—51
8Ra	....	....	....	—18	—18	—54
9Ra	18	—14	—23	—28	—37	—64
1T	284	209	222	276	249	219
2T	....	135	175	151	145	195
3T	110	87	78	92	86	94
4T	83	64	....	89	52	75

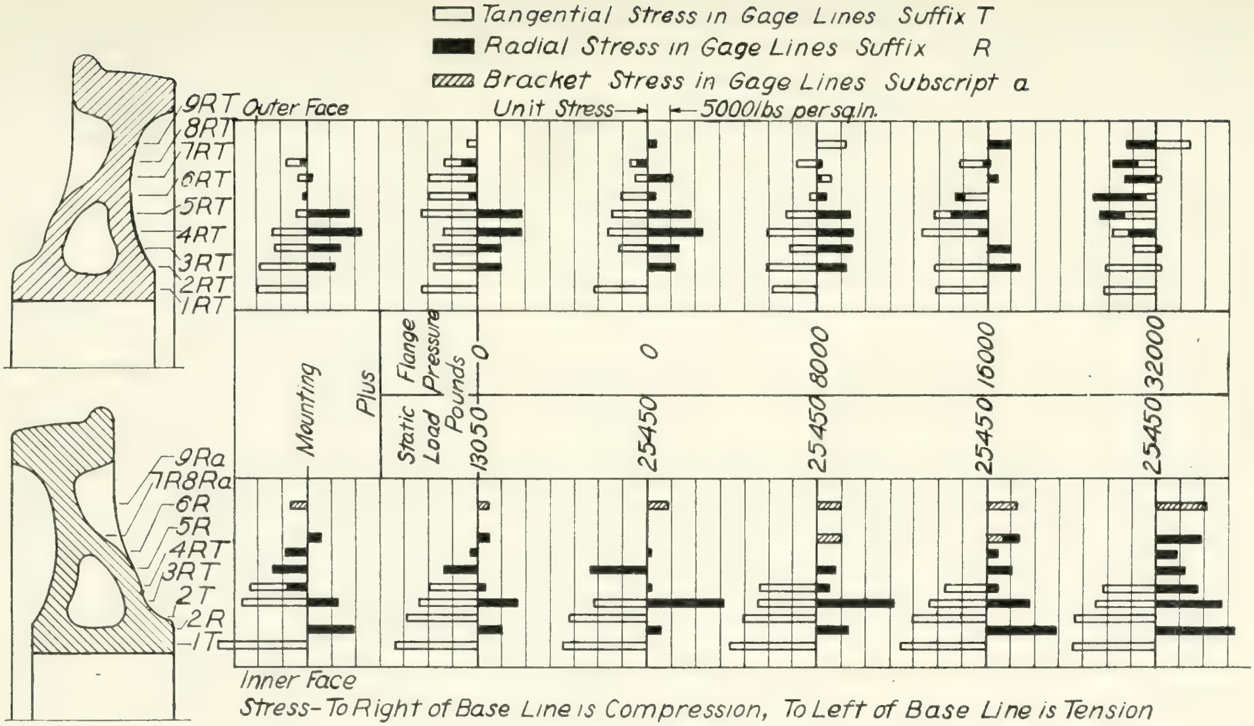


FIG. 5. CORRESPONDING SIMPLE STRESSES IN 33-IN 840-LB. ARCH PLATE WHEEL NO. 942 805 CAUSED BY MOUNTING, STATIC LOAD, AND FLANGE PRESSURE

On the outer face of the wheel the maximum corresponding simple tensile stress due to mounting was 10 000 lb. per sq. in. and was in a tangential direction at the bore. As the distance from the bore increased, the stresses in the tangential direction decreased up to the point where the inner and outer plates join, beyond which a slight increase occurred. In the radial direction on the outer face the maximum compressive strain due to mounting corresponded to a simple stress of 11 600 lb. per sq. in., and occurred at gage-line 4R. The variation in the magnitude of the stresses in the radial direction clearly indicated the presence of bending in addition to the direct thrust due to mounting.

The maximum tensile strain on the inner face due to mounting was likewise in a tangential direction at the bore, and corresponded to a simple stress of 18 400 lb. per sq. in. The magnitude of the tensile stresses in the tangential direction decreased as the distance from the bore increased. The stresses in the radial direction on the inner face varied from 9600 lb. per sq. in. compression at the gage-line nearest the bore to zero between gage-lines 3R and 4R. The stress then became tensile at 4R, reaching a maximum of 7000 lb. per sq. in. at 5R, beyond which the tension decreased and the stress again became compressive at 7R. This variation in stress in the radial lines also indicated the presence of bending in addition to the direct thrust due to mounting.

9. *Strains and Corresponding Simple Stresses Due to Combined Effect of Mounting and Static Load.*- Reference to Table 1 and Fig. 5 shows that upon the application of a 13 050-lb. static load the tensile strains on the outer face at the tangential gage-lines 1T to 4T inclusive were not materially different from those caused by forcing the wheel on the axle. The effect of the load was indicated, however, by an increased tension at tangential gage-lines 5T to 9T inclusive. In the radial direction the compressive strains, although different from those caused by mounting, were relatively small. On the inner face the tensile strains at the tangential gage-lines 1T and 4T inclusive were somewhat smaller than those due to mounting alone. In the radial direction the compressive strains on the inner face were also of small value.

With the exception of those at gage-lines 5T and 6T, the strains in the tangential direction on the outer face, due to the combined effect of mounting and a 25 450-lb. static load, were not materially different from those caused by mounting alone. At 5T and 6T the static load increased the corresponding simple stresses 5200 and 4600 lb. per sq. in., respectively. At the remaining tangential gage-lines the change due to the application of the static load was never in excess of 1300 lb. per sq. in. In the radial direction the greatest change occurred at gage-line 7R, where the compression was increased 3800 lb. per sq. in. On the inner face no material change occurred at the tangential gage-lines due to the application of the 25 450-lb. static load. The stresses at the radial gage-lines, however, differed considerably from those due to mounting alone. The most significant change was at radial gage-line 5R, at which the tension of 7000 lb. per sq. in. due to mounting was increased to 11 700 lb. per sq. in. by the application of the 25 450-lb. load. This increase, however, is not of serious import for two reasons: first, the static load is practically the maximum that the wheel should carry and second, the imposition of flange pressure decreased the tension until at 32 000-lb. flange pressure a compressive stress of 6300 lb. per sq. in. existed at that gage-line.

10. *Strains and Corresponding Simple Stresses Due to Combined Effect of Mounting, Static Load, and Flange Pressure.*—The combined effect of mounting, 25 450-lb. static load, and 32 000-lb. flange pressure as shown by Table 1 and Fig. 5 produced tension at all gage-lines, both radial and tangential, on the outer face of the wheel, except 2R, 3R and 9T. The significant fact is that the combination of stress-producing factors produced tension in the radial direction from 4R to 9R. Some later tests showed that prolonged brake application in the case of the Arch Plate type of wheel, also produced tension in the radial direction, and further, that the maximum tensile stress caused by brake application may be expected to occur in the region 4R to 6R, or near the junction of the inner and outer plates. In themselves the stresses here noted are not dangerous. If, however, to these stresses there is added the effect of a severe and prolonged brake-shoe application, the stresses might become of such magnitude as to cause incipient cracks probably leading to wheel failure. On the inner face of the wheel the combined effect of mounting, static load, and flange pressure gave rise to stresses in a tangential direction practically equal to those due to mounting alone. There was a difference, however, in the

distribution of the stresses in the radial direction. In mounting, the stresses at 4R, 5R and 6R were tensile, but these were changed to compressive stresses by the addition of static load and flange pressure. The compressive stresses found at the radial gage-lines on the inner face were relatively unimportant. The application of either the 8 000-lb. or the 16 000-lb. flange pressure, in combination with the 25 450-lb. static load, produced an effect intermediate between that due to the static load alone and that due to the combination of the static load with the 32 000-lb. flange pressure.

11. *Remarks on Tests.*—The compressive stresses found for the various load conditions were relatively unimportant. The maximum corresponding simple tensile stress (18 400 lb. per sq. in.) was due to mounting and was found at the gage-lines nearest the hub on the inner face. The application of static load and flange pressure did not materially affect this stress. Stresses of such high magnitude occurring in the hub regions are generally less dangerous than stresses of lesser value at other positions in the wheel. This is due to the fact that the stress in the hub region is a steady one, that is, the material is not subjected to repeated stress. The combined effect of mounting and the application of a maximum static load and a flange pressure probably in excess of that encountered in normal wheel service was found to produce tension (with a few minor exceptions) in both radial and tangential directions on the outer face of the 840-lb. Arch Plate wheel. On the inner face over the region investigated, this combination of loading was found to produce tension at the tangential and compression at the radial gage-lines. The tensile stresses on the outer face near the junction of the inner and outer plates resulting from the application of maximum static load and flange pressure are subject to repetition and at some points to complete reversal. The number of such reversals of stress that would occur in the normal life of a wheel, due to the occurrence of a sufficiently high flange pressure caused by traversing curves of small enough radius at high enough speed, is relatively small compared with the millions of reversals found necessary to cause failure under similar conditions in investigations of repeated stress. Hence, the question of severity of stress due to the combined effect of mounting, static load, and flange pressure may be considered from the static viewpoint alone. It does not appear

that the stresses due to this combination of loading are severe in themselves, but if these stresses are increased by abuse of the wheel, through prolonged brake application, they might become large enough to cause incipient cracks leading to wheel failure.

V. DETERMINATION OF ULTIMATE STRENGTH OF FLANGE OF CHILLED IRON CAR WHEEL

12. *Apparatus Used in Making Tests of Ultimate Strength of Flanges.* A special piece of testing apparatus, consisting essentially of a steel beam built up of channels and plates, used in determining the ultimate strength of the flanges of chilled iron car wheels, is shown in Fig. 6. This beam was placed on the weighing table of a 600 000-lb. testing machine, and the wheel to be tested was supported thereon, as shown in Fig. 6, at opposite points on a diameter, at one side by a piece of tool steel pressing against the flange, and at the other by a semi-circular piece of steel supporting the tread. The location of the point of application of the load on the flange could be varied by means of a block and wedge applied to the tread at a point diametrically opposite. This block and wedge further acted to take the horizontal pressure component due to the angularity of the flange at the point of load application. The load was applied to the wheel, as shown, through a block with a spherical seat. From three to six tests to fracture were made on each wheel.

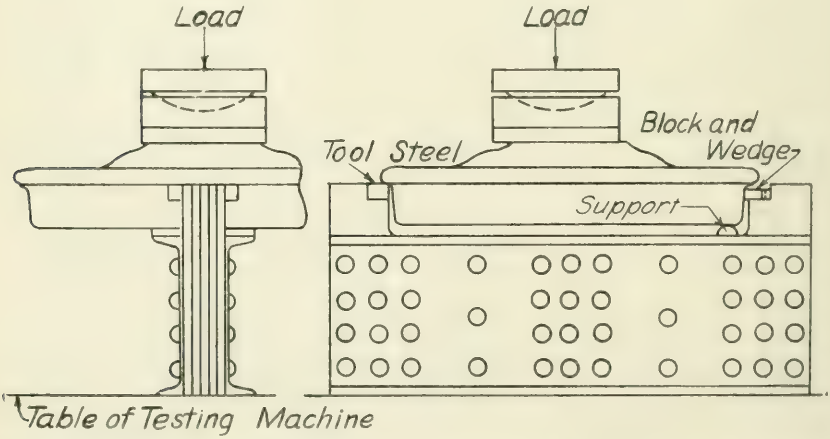


FIG. 6. APPARATUS USED TO DETERMINE ULTIMATE STRENGTH OF FLANGES

13. *Shapes of Flanges Tested and Test Conditions.*—In the case of a number of the wheels tested the flanges were ground to represent various stages of wear, so that the ultimate strengths determined in these cases furnish data as to the safety of wheels similarly worn in service. The location of the point at which the pressure was applied to the flange was also varied, and thus data were secured applicable to the case of wheels operating on improperly maintained track, on which the flanges may strike frogs, switches, etc. The various contours of the flanges tested, and for each case the point of application of the load, the component of the ultimate strength of the flange in a direction parallel to the bore, and the direction of fracture are shown in Figs. 7 to 40 inclusive. The dotted lines appearing on some of the figures indicate the shape of the section between two adjacent brackets and, further, that the break occurred in the region of a bracket, the actual contour at the point of fracture being shown by the full lines. The tests represented by Figs. 7 to 40 inclusive were made with a view of determining to what extent the ultimate strength of the flange was affected by variations in the flange thickness and the position at which the flange pressure was applied. Irregularities occurred, however, in the results of these tests which preclude correct deduction concerning the effect of flange thickness. The irregularities in the results were caused by differences in the various wheels with respect to depth of chill, chemical composition, physical properties of the metal, etc. In order to obtain information of value regarding the effect of additional metal in the flange region, an attempt was made to eliminate these differences by using wheels cast from special patterns, in which one-half of the circumference had the standard M. C. B. flange and the other half had additional metal on the inner face. Three types of wheels, represented by Figs. 30 to 40 inclusive, were thus cast and from these the effects in both new and worn wheels of, first, increasing the thickness of the flange, second, increasing the thickness of the tread, and, third, increasing the thickness of both flange and tread simultaneously, were determined. To make the comparisons still more direct the loads were in every case applied to the flanges of these specially prepared wheels on a radius midway between the points where two adjacent brackets entered the tread, thus eliminating as far as possible the complex effect of the support of the brackets. In addition, the flange pressures were in each case applied at identical

distances below the tread, thereby removing the effect of difference in the moment of the load. Because of these precautions it is felt that the results obtained from these wheels cast from specially prepared patterns are truly representative.

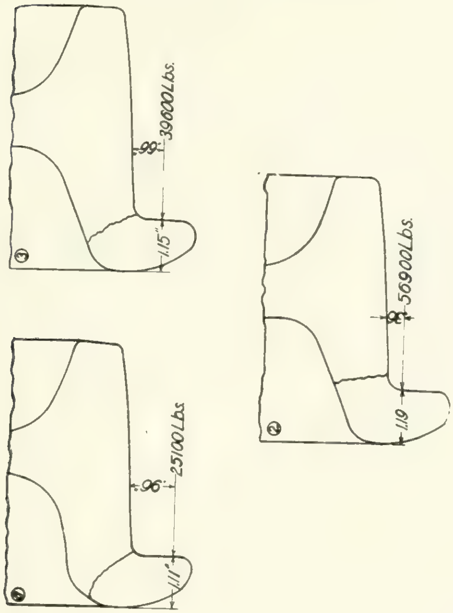


FIG. 8. 625-LB. M. C. B. WHEEL NO. 737 425. THROAT GROUND TO REPRESENT WORN FLANGE

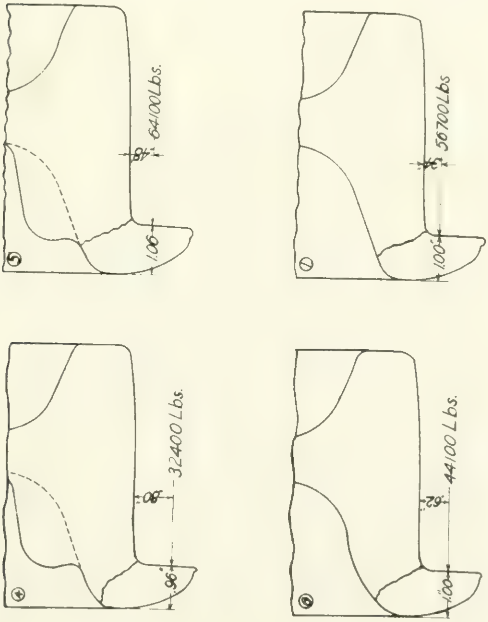


FIG. 7. 625-LB. M. C. B. WHEEL NO. 737 401. THROAT GROUND TO REPRESENT WORN FLANGE

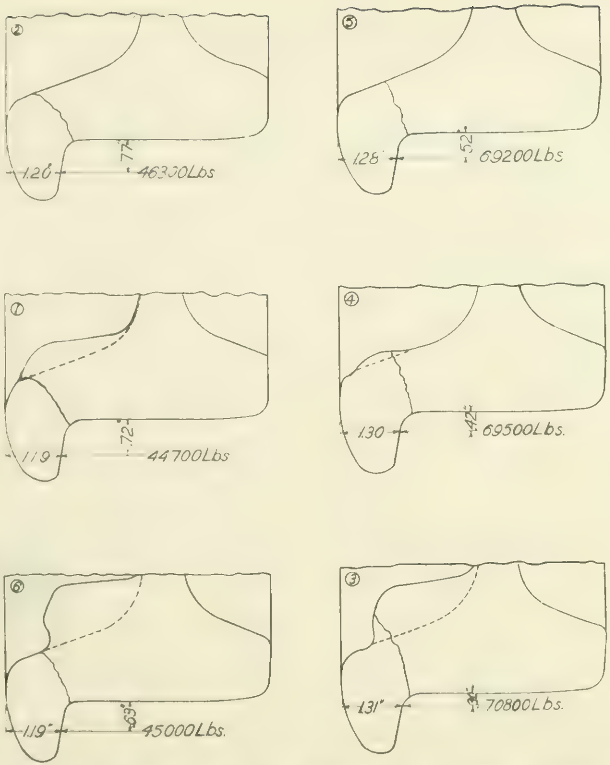


FIG. 9. 625-LB. M. C. B. WHEEL NO. 737 474. THROAT GROUND TO REPRESENT WORN FLANGE

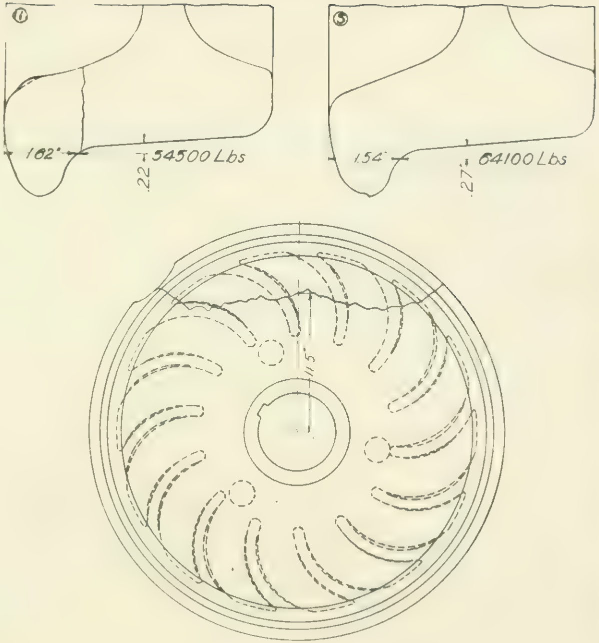


FIG. 10. 625-LB. M. C. B. WHEEL NO. 632 680. FLANGE AS CAST

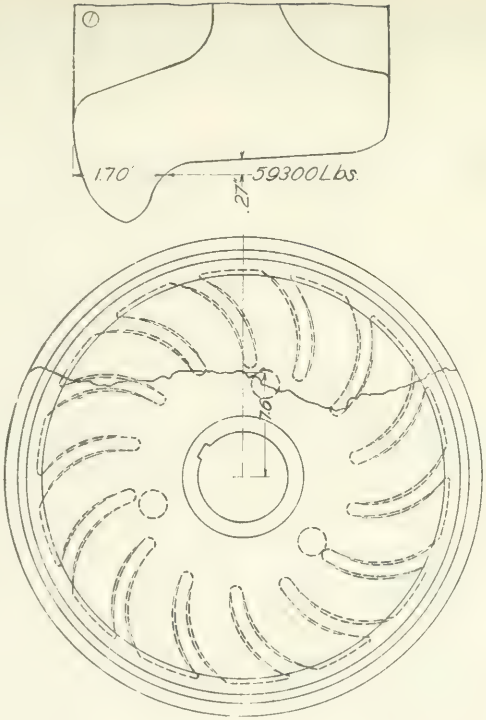


FIG. 11. 650-LB. M. C. B. WHEEL NO. 633 452.
FLANGE AS CAST

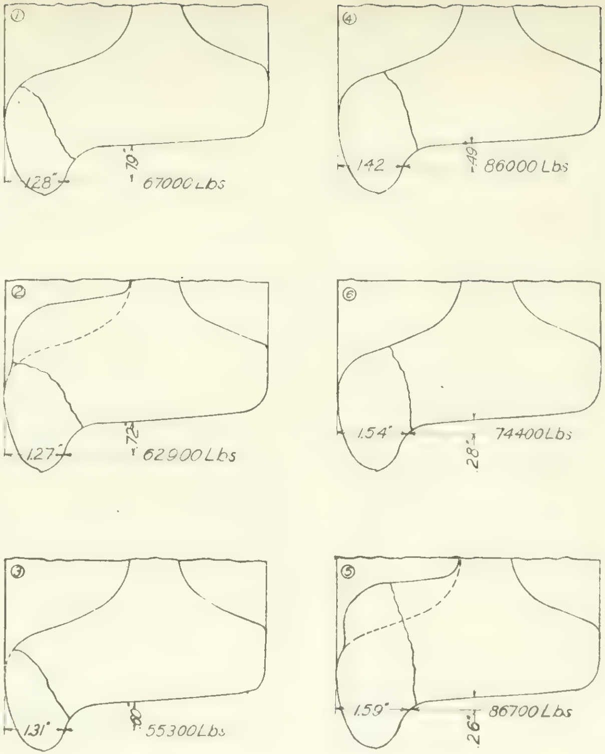


FIG. 12. 675 LB. M. C. B. WHEEL NO. 632 679.
FLANGE AS CAST

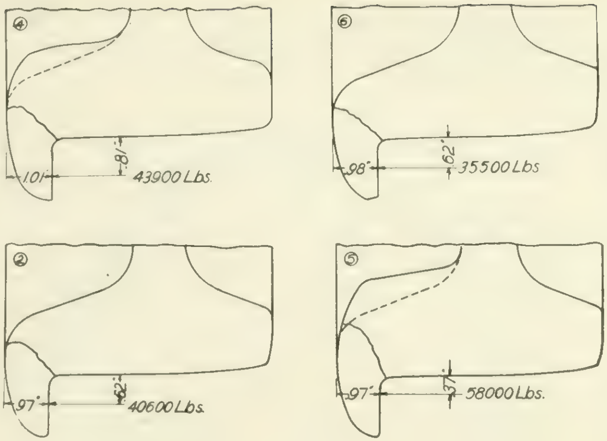


FIG. 13. 725-LB. M. C. B. WHEEL NO. 738 957.
THROAT GROUND TO REPRESENT WORN FLANGE

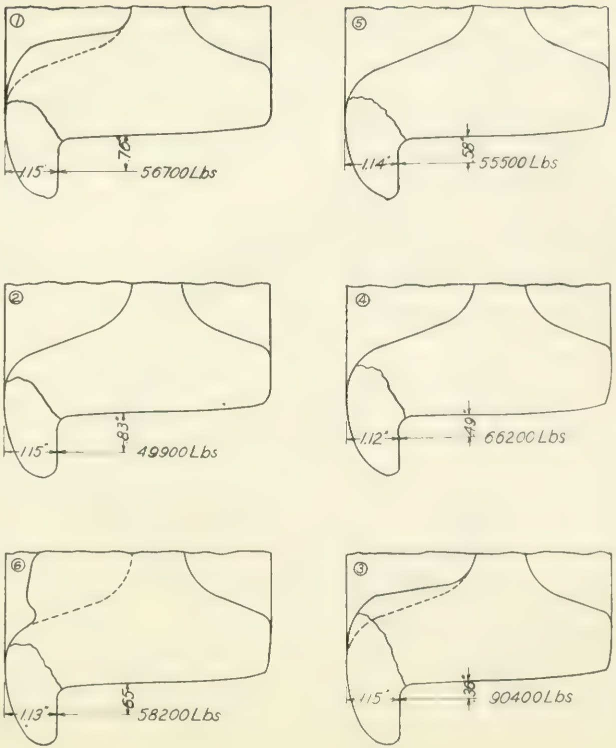


FIG. 14. 725-LB. M. C. B. WHEEL NO. 734 502.
THROAT GROUND TO REPRESENT WORN FLANGE

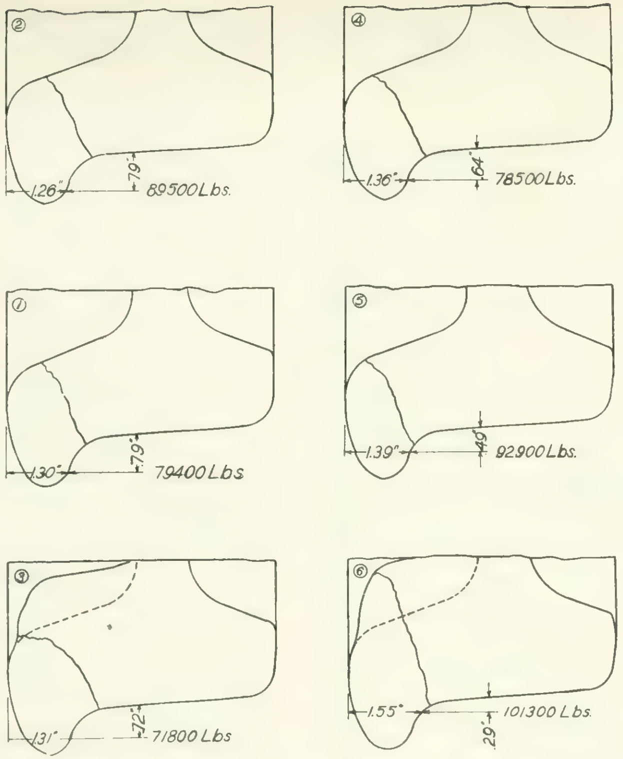


FIG. 15. 725-LB. M. C. B. WHEEL NO. 671 449.
FLANGE AS CAST

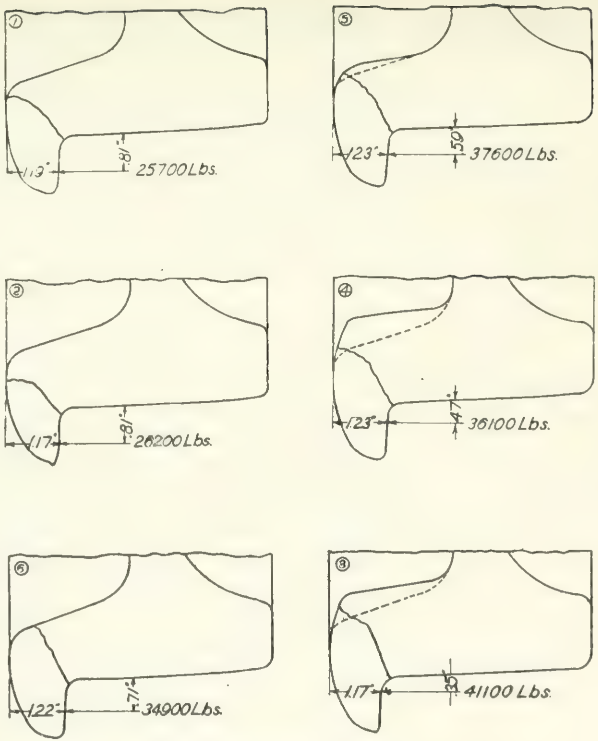


FIG. 16. 740-LB. ARCH PLATE WHEEL NO. 04 844.
THROAT GROUND TO REPRESENT WORN FLANGE

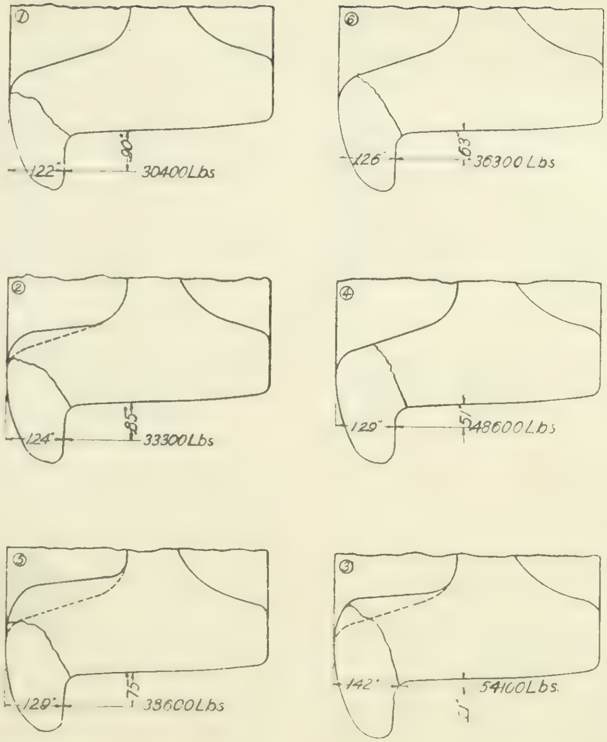


FIG. 17. 740-LB. ARCH PLATE WHEEL NO. 04 836. THROAT GROUND TO REPRESENT WORN FLANGE

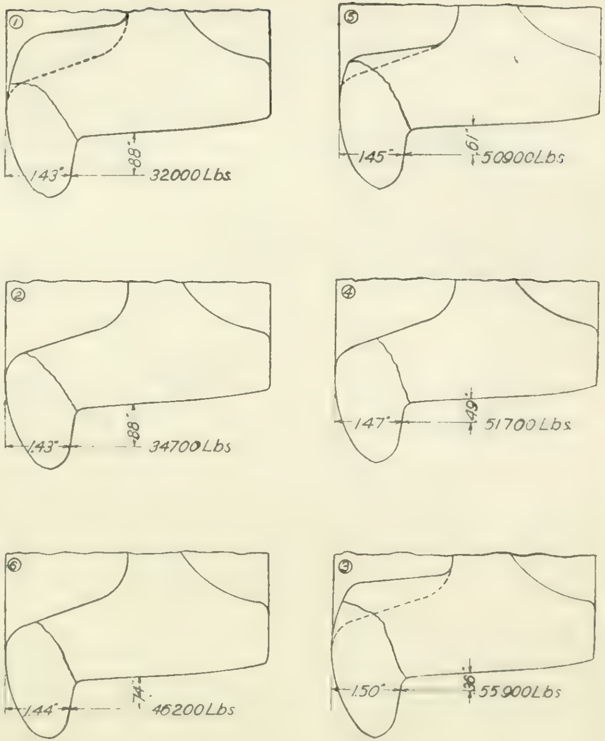


FIG. 18. 740-LB. ARCH PLATE WHEEL NO. 04 847. THROAT GROUND TO REPRESENT WORN FLANGE

VI. Results of Tests

14. *General Remarks on Tests.*—A wide diversity of opinion prevails as to the magnitude of flange pressures and their importance. As previously explained, attempts to calculate accurately the flange pressures occurring in service are unsatisfactory on account of the many variables which enter into the problem. Therefore, no attempt has been made to estimate factors of safety for the various sections broken. Furthermore, the comparisons made herein are confined to flange fractures or breaks made on the same wheel in order to exclude as far as possible the effect of outside factors, such as depth of chill, character of metal, etc. As a consequence the comparisons given may be deemed to be fairly representative of the factors considered. The results of these tests are given in Figs. 7 to 40 inclusive, which are followed by Figs. 41 to 46 inclusive, reproduced from photographs of the several wheels taken after the breaking of the flanges. The photographic views are presented mainly to show the location, with respect to the brackets, at which the loads were applied. They also show something of the nature and shape of the fracture and by a study of them the approximate size of the piece broken out can also be estimated, since the maximum diameter of the flange is 35 inches. Since the flange was broken at several places on each wheel, each fracture was given an identifying number. In Figs. 7 to 40 this number will be found in the upper left hand corner of each of the small section figures, while the corresponding numeral appears on the photographic view in the vicinity of the fracture.

15. *Effect of Varying Position at which Pressure Is Applied to Flange.*—In the tests illustrated by Figs. 7 to 27 inclusive, and photographic Figs. 41 to 44 inclusive, it was found that, in general, if the side thrusts were applied to the flange on a radius *midway* between the points at which two adjacent brackets enter the tread, the flange pressure necessary to produce failure in the case of a wheel with *worn* flanges was an inverse function of the distance of the point of pressure application from the tread; that is, the greater the distance the smaller

the ultimate strength of the flange. This is shown in Table 2 compiled from results taken from Fig. 17.

TABLE 2

RELATION BETWEEN POINT OF APPLICATION OF LOAD AND STRENGTH OF FLANGE

Wheel No. 04 836

Fracture No.	Thickness of Flange in.	Distance of Point of Application Below Tread in.	Pressure Required to Cause Fracture lb.
1	1.22	0.90	30 400
6	1.26	0.63	36 300
4	1.29	0.51	48 600

TABLE 3

RELATION BETWEEN POINT OF APPLICATION OF LOAD AND STRENGTH OF FLANGE

Wheel No.	Fracture No.	Thickness of Flange in.	Distance of Point of Application Below Tread in.	Pressure Required to Cause Fracture lb.
04 476	3	1.49	0.61	86 300
	5	1.52	0.49	63 300
942 682	5	1.76	0.55	95 600
	3	1.71	0.28	79 000
107 348*	1	1.53	1.26	109 600
	3	1.63	1.09	90 500
103 219	2	1.63	0.82	92 000
	4	1.72	0.60	88 300

* After having fulfilled the guaranteed mileage, this wheel was withdrawn from service, on account of being tread-worn. The flange had the same general shape as that of a new wheel.

A result of this kind was to be expected and was obtained in all the tests on worn wheels in which the thrust was applied at the mid-position between two adjacent brackets. The condition that the greater the distance from tread to point of load application, the less the ultimate strength of flange, was not found to hold, however, in the case of flanges not worn or only slightly worn as is shown in Table 3 compiled from results taken from Figs. 19, 21, 23, and 25.

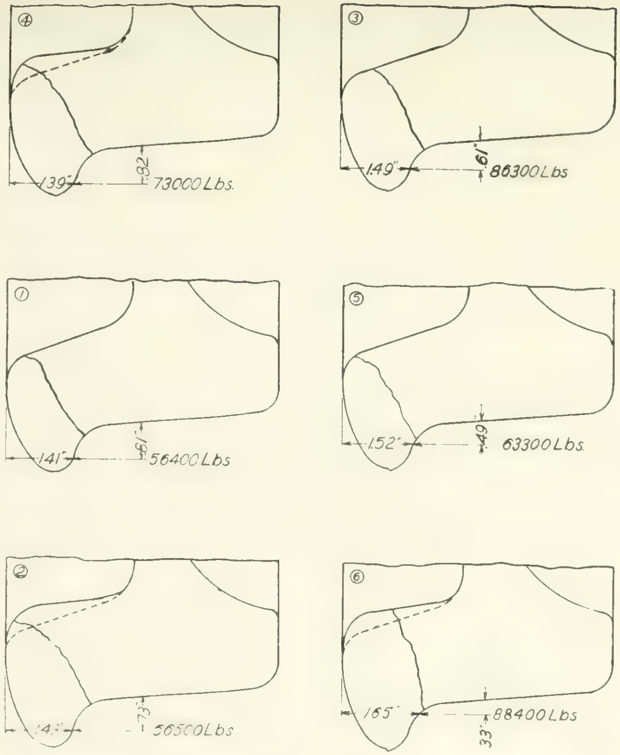


FIG. 19. 740-LB. ARCH PLATE WHEEL NO. 04 476. FLANGE AS CAST

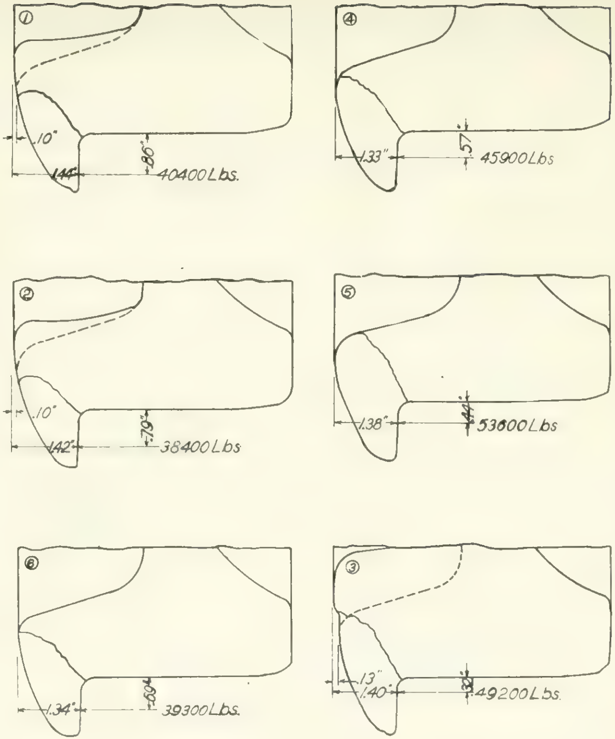


FIG. 20. 840-LB. ARCH PLATE WHEEL NO. 942 674. THROAT GROUND TO REPRESENT WORN FLANGE

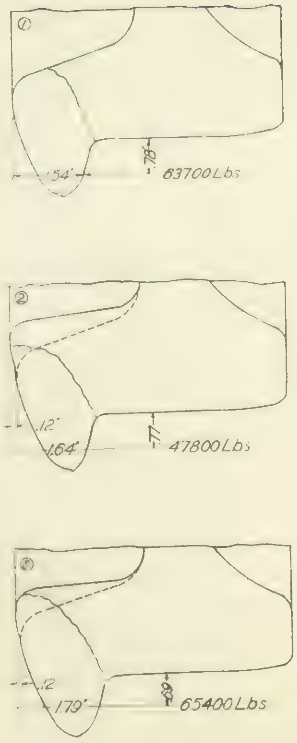


FIG. 21. 840-LB. ARCH PLATE WHEEL NO. 942 682. THROAT GROUND TO REPRESENT WORN FLANGE

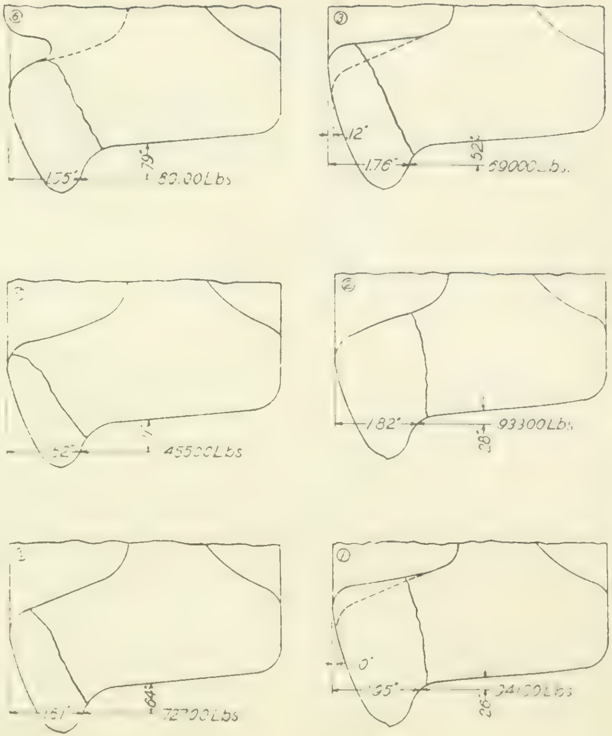


FIG. 22. 850-LB. ARCH PLATE WHEEL NO. 659 232. FLANGE AS CAST

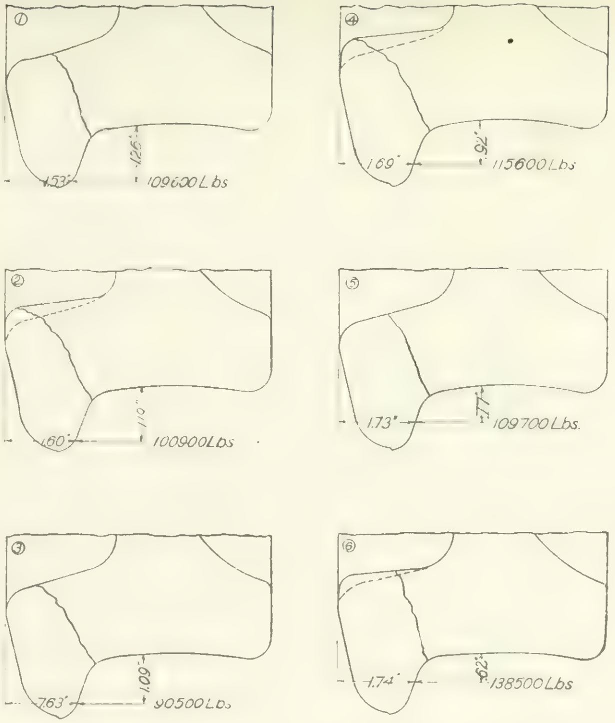


FIG. 23. 850-LB. ARCH PLATE WHEEL NO. 107 348. WHEEL REMOVED FROM SERVICE

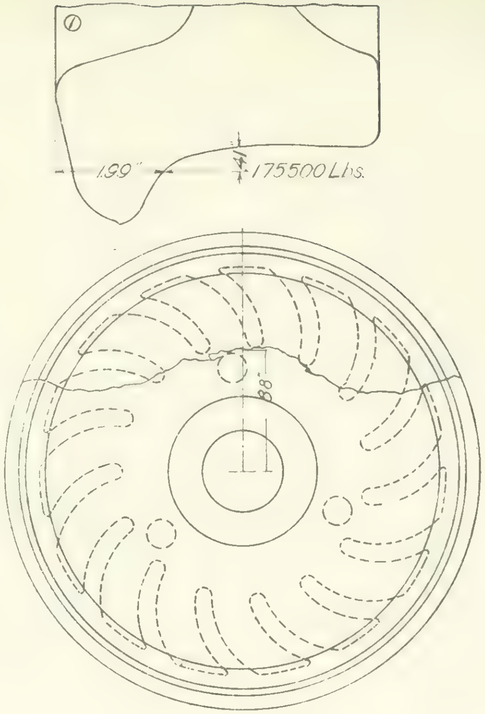


FIG. 24. 850-LB. ARCH PLATE WHEEL NO. 107 346. WHEEL REMOVED FROM SERVICE

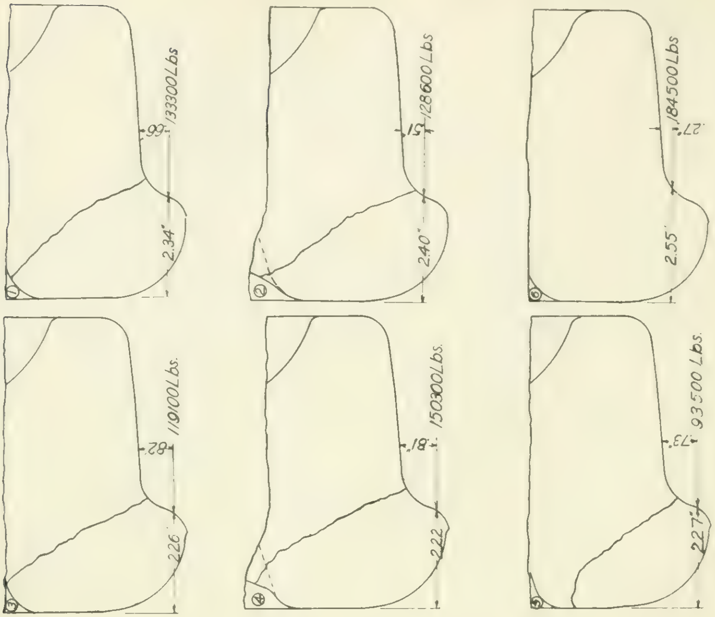


FIG. 26. 960-LB. ARCH PLATE WHEEL NO. 659 338.
EXTRA HEAVY FLANGE

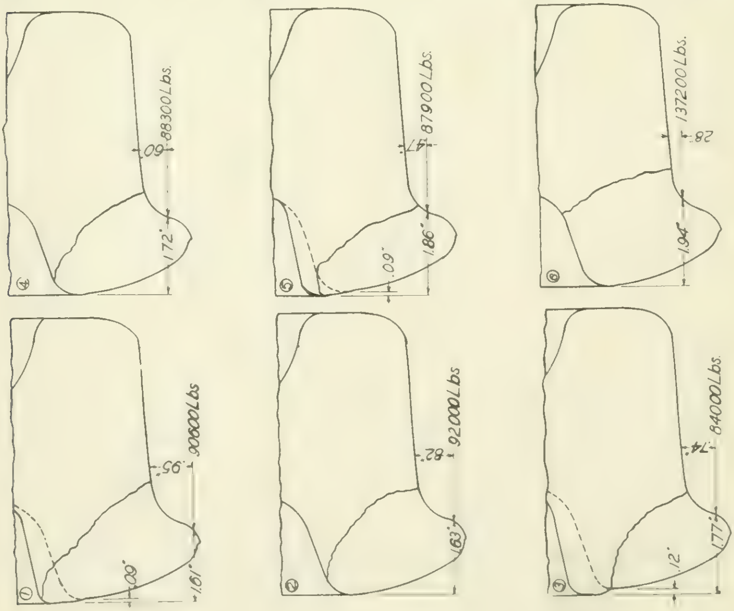


FIG. 25. 950-LB. ARCH PLATE WHEEL NO. 103 219.
FLANGE AS CAST

Another instance in which the inverse relation between distance from tread and ultimate strength was found not to hold was that where the pressure was applied to the flange on the same radius as that on which a bracket enters the tread. In this case there were present in the results certain variations which indicated a complex supporting effect due to the bracket.

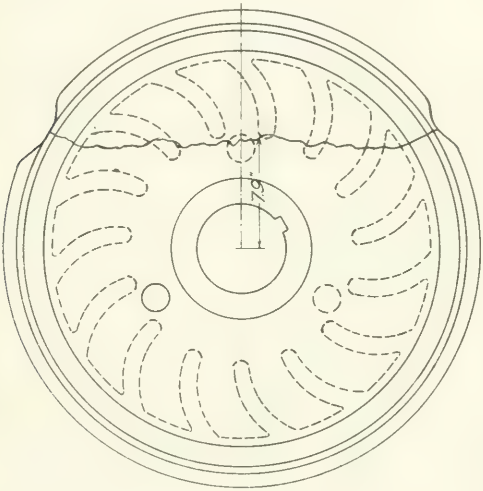


Fig. 27. 960-lb. Arch Plate Wheel No. 659 338 Broken Through Plate by Flange Pressure

Under ordinary service conditions the rail pressure is applied to the flange at a small distance below the tread and those cases of fracture occurring in the tests in which the load was applied closest to the tread are most representative of the type of flange failure to be expected in service. Similarly, the ultimate strengths determined in tests where the load was close to the tread are more representative of the flange strength of the wheel in normal service than those found when the load was applied farther from the tread.

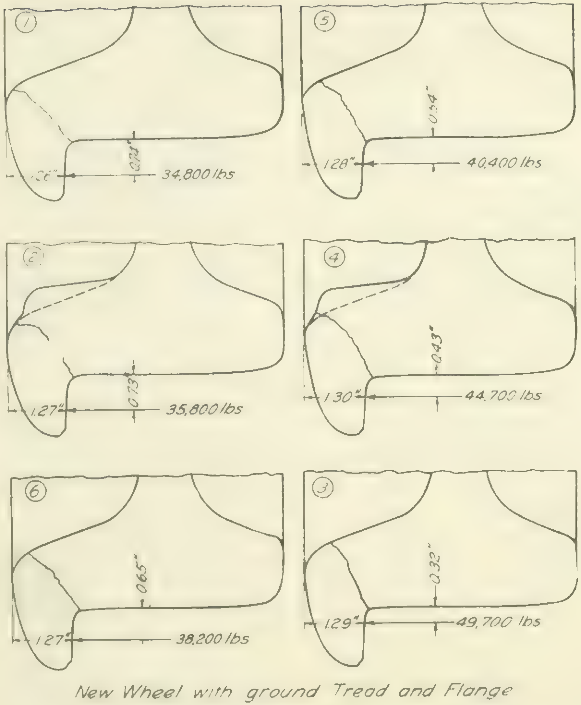


FIG. 28. 725-LB. M. C. B. WHEEL NO. 462 442. FLANGE PROPOSED BY A. M. C. C. W. THROAT GROUND TO REPRESENT WORN FLANGE

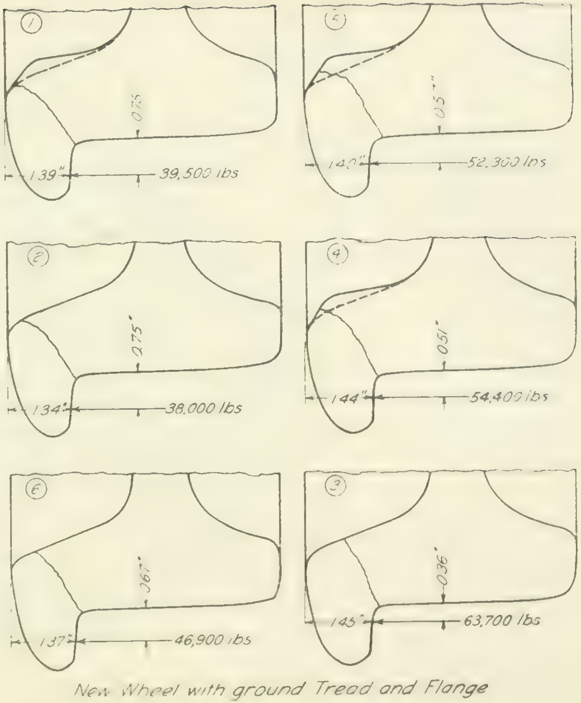


FIG. 29. 725-LB. M. C. B. WHEEL NO. 462 441. FLANGE PROPOSED BY A. M. C. C. W. THROAT GROUND TO REPRESENT WORN FLANGE

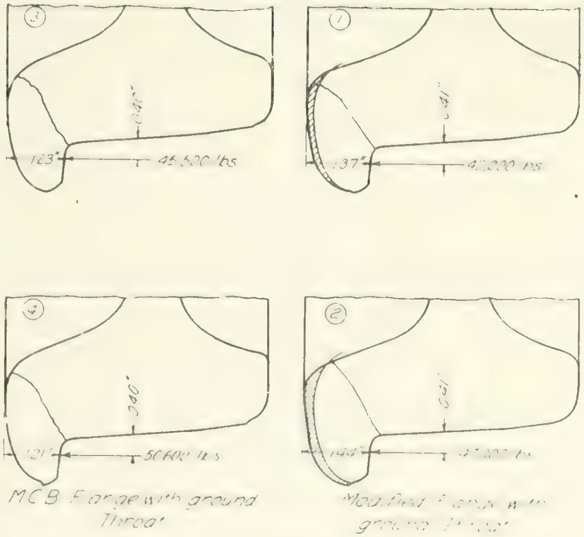

③
.040"
.123"
45,500 lbs
①
.041"
.137"
④
.040"
.121"
50,600 lbs
②
.041"
MCB Flange with ground Throat
Modified Flange with ground Throat

FIG. 30. 725-LB. M. C. B. WHEEL NO. 158 505. THROAT GROUND TO REPRESENT WORN FLANGE

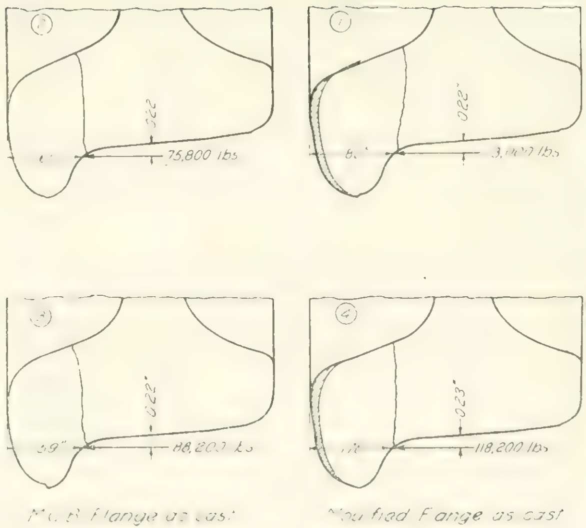


FIG. 31. 725-LB. M. C. B. WHEEL NO. 158 506. FLANGE AS CAST

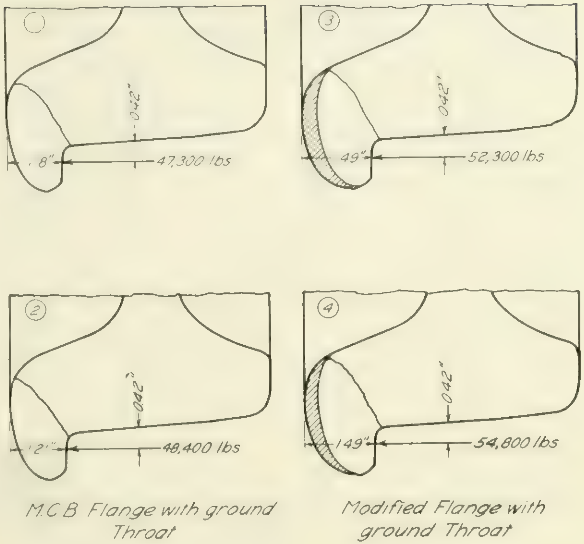


FIG. 32. 725-LB. M. C. B. WHEEL NO. 158 511. THROAT GROUND TO REPRESENT WORN FLANGE

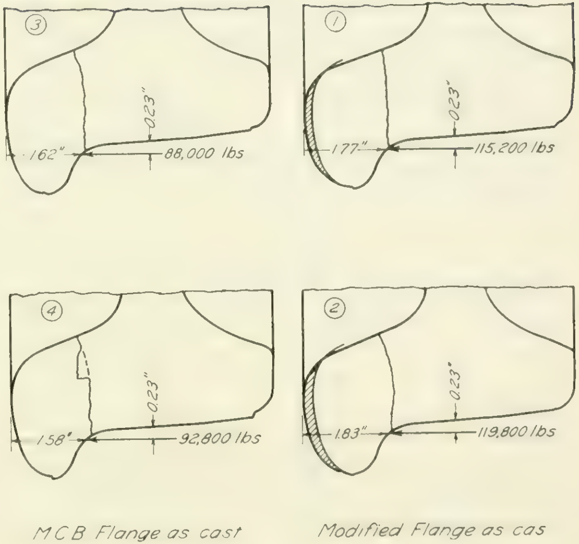
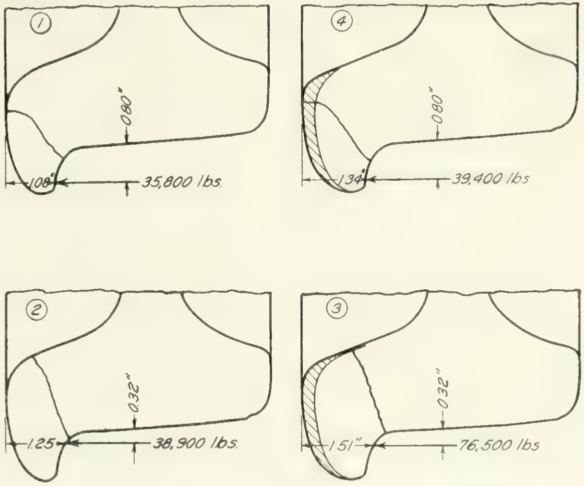


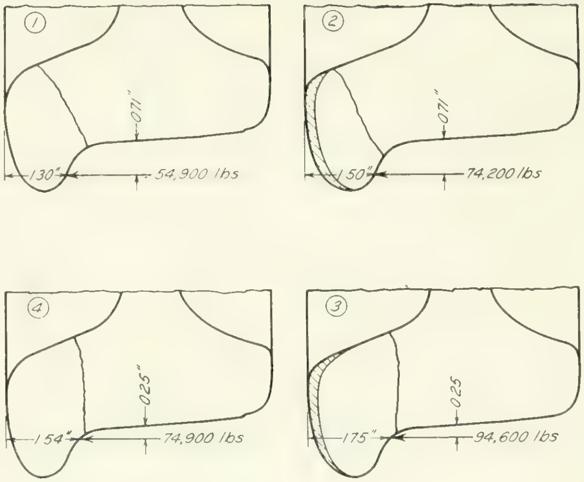
FIG. 33. 725-LB. M. C. B. WHEEL NO. 158 510. FLANGE AS CAST



M.C.B Flange with ground Throat

A.M.C.C.W. Flange with ground Throat

FIG. 34. 725-LB. M. C. B. WHEEL No. 132 648. THROAT GROUND TO REPRESENT WORN FLANGE

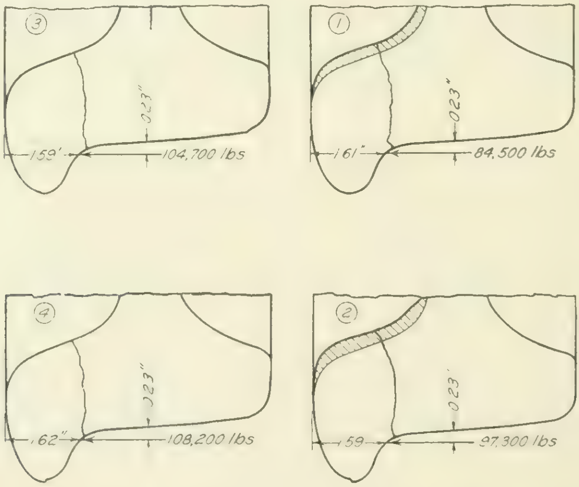


M C B Flange as cast

A M C C W Flange as cast

FIG. 35. 725-LB. M. C. B. WHEEL No. 132 649. FLANGE AS CAST

16. *Effect of Thickening Flange.*—In Figs. 28 and 29 and photographic Figs. 44 and 45 are shown the results of tests made on two wheels having the flange proposed by the Association of Manufacturers of Chilled Car Wheels to replace the present type of flange.* In both wheels the throats were ground to represent worn flanges. The tests made thereon determined the ultimate strength when the load was applied at various points on the flange but, as already stated, on account of probable differences in the physical properties of the wheels, the results obtained were not considered as conclusively showing the effect of the additional metal. To show this effect in a more reliable manner a special pattern was made in which one-half of the circumference had the present standard flange and the other half had the proposed flange. Six wheels of this type were cast, of which three were tested after being ground to represent worn flanges and three



MCB Flange as cast

Modified Flange as cast

Fig. 36. Sections of Car Wheels Showing Pressure Required to Break Flange; Point of Load Application; Direction of Fracture; and Variation in Ultimate Strength Due to Thickening Tread.
725 lb. M. C. B. Wheel No. 158 525. Flange as Cast

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* See first footnote page 10.

were tested with the flanges as cast. Figs. 30 to 35 inclusive show the test conditions and results on wheels prepared in this manner. As arranged in the figures, any two sections on a horizontal line are directly comparable with respect to difference in the ultimate strength existing between the proposed A. M. C. C. W. and the standard M. C. B. flanges. In the figures with the cross-hatching the outer line of the drawing is the contour of the proposed A. M. C. C. W. section at the point where the pressure was applied, while the inner line of the cross-hatching shows the standard M. C. B. section. The wheel sections without cross-hatching are the standard M. C. B. sections.

Table 4 gives results taken from the tests represented by Figs. 30 to 35 inclusive, and also the ratio of the ultimate strength of the proposed A. M. C. C. W. flange to the ultimate strength of the standard M. C. B. flange.

With the exception of the first wheel, No. 158 505, mentioned in the table, the proposed flange was in each case stronger than the present standard flange. For wheel No. 158 505 the outer diameters of the

Table 4

Comparative Strength of Standard M. C. B. Flange and Proposed A. M. C. C. W. Flange

Wheel No	Distance of Point of Application Below Tread in.	Ultimate Strength of Standard M. C. B. Flange lb.	Ultimate Strength of Proposed A. M. C. C. W. Flange lb.	Ratio of Strength Proposed to Standard
158 505	0.40	45 500	42 200	0.93
(Worn Flange)	0.40	50 600	47 000	0.93
158 511	0.42	47 200	52 300	1.10
(Worn Flange)	0.42	48 400	54 800	1.13
132 648	0.80	35 800	39 400	1.10
(Worn Flange)	0.32	38 900	76 500	1.96
158 506	0.22	75 800	113 000	1.49
(Flange as cast)	0.22	88 200	118 200	1.34
158 510	0.23	88 000	115 200	1.31
(Flange as cast)	0.23	92 800	119 800	1.29
132 649	0.71	54 900	74 200	1.35
(Flange as cast)	0.25	74 900	94 600	1.26

flanges in the contours numbered 1, 2, and 4 show that the chiller had not been concentric with the mold when this wheel was cast. This eccentricity resulted in increasing the thickness of the tread on the half having the standard flange and decreasing the corresponding thickness on the half having the proposed flange, the difference in thickness being about 1/8 inch. This condition is evident in the figures showing breaks Nos. 1 and 2 in which the contour lines for the proposed and the standard flanges are seen to intersect on the under side of the tread, and also may be seen by a study of the shapes of the tips of the flanges in the figures showing fractures Nos. 1, 2, and 4. A study of all the data obtained with reference to the ultimate strength of flanges would apparently lead to the conclusion that a certain relation or ratio between the thickness of the tread and that of the flange can be found which corresponds to a maximum flange strength for a particular thickness of flange or tread. Assuming the existence of such a relation, it follows that the addition of metal to the flange or tread alone, but not to both flange and tread, may result not only in no further increase in flange strength but even in a decreased flange strength. It is probable that the greater thickness of the tread of that part of wheel No. 158 505, having the standard flange, accounts for the greater ultimate strength of the flange in this case, in that the ratio of flange to tread thickness was more nearly the ideal in the case of the standard flange portion of the wheel than it was in the case of the proposed flange portion of the wheel.

If on account of the unusual flange and tread proportions due to the misplaced chiller the data for wheel No. 158 505 be excluded in considering the differences between the standard and proposed flanges, the results from the remaining five wheels listed in the table show the ultimate strength of the proposed flange to be from 10 per cent to 96 per cent greater than that of the standard flange. This table also shows that in the case of unworn flanges the advantage in favor of the proposed flange ranges from 26 per cent to 49 per cent. These increases in strength were obtained by the addition of 15 lb. of metal to the wheel. Since these specially prepared wheels were cast from 725-lb. M. C. B. patterns, the added metal represented 2.1 per cent of the weight of the wheel. From this it may be concluded that 2.1 per cent of metal added to this type of wheel, as suggested by the Association of Manufacturers of Chilled Car Wheels, would increase the ultimate strength of the present standard M. C. B. flange while new, roughly, from 26 per cent to 49 per cent. As the wheel approaches the condemning limits through flange wear, the advantage of the proposed increase in thickness is apparently somewhat less than for new flanges. The results for the two worn wheels considered indicate a gain in strength of from 10 per cent to 13 per cent in 3 cases and of 96 per cent in one case resulting from the thicker flanges. It is likely that similar results would be noted if tests were made on M. C. B. type wheels of different weights or on Arch Plate type wheels. Variations in thickness of the tread and in the support given to the tread might, however, in the case of differently proportioned wheels, produce results more or less at variance with those here presented. Additional experiments with other types of wheels would be desirable.

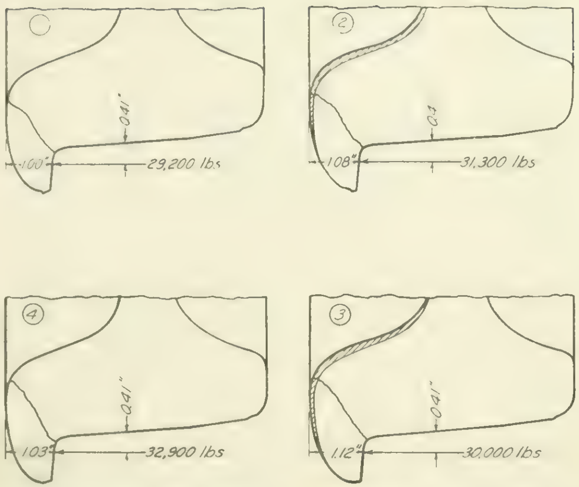
17. *Effect of Thickening Tread.*—To show the effect of thickening the tread without thickening the flange, one wheel, No. 158 525, shown in Figs. 36 and 46, was cast in which half the circumference had the standard M. C. B. thickness of tread and the other half a thickness of tread about 3 16 inch greater. In Fig. 36 the cross-hatching indicates the approximate amount and location of the additional metal. The data obtained from this wheel are given in Table 5.

Only one wheel was tested with metal added in this manner and the results cannot, therefore, be considered as conclusive. If further

TABLE 5

COMPARATIVE STRENGTH OF STANDARD M. C. B. FLANGE AND FLANGE OF WHEEL WITH THICKENED TREAD

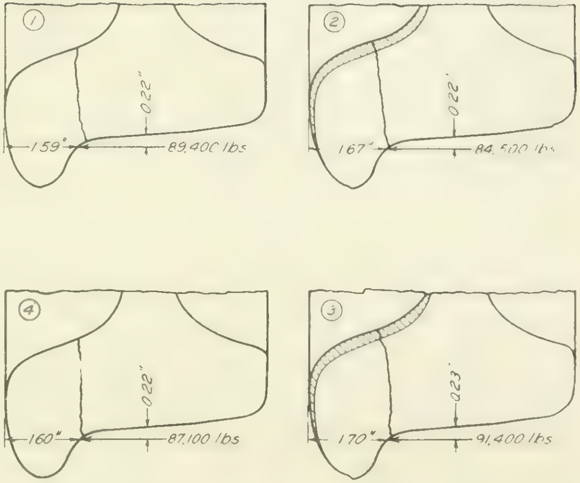
Wheel No.	Distance of Point of Application Below Tread in.	Ultimate Strength of Standard M. C. B. Flange lb.	Ultimate Strength of Flange of Wheel With Tread Thickened lb.	Ratio of Strength Thickened to Standard
158 525	0.23	104 700	84 500	0.81
(Flange as cast)	0.23	108 200	97 300	0.90
Average		106 450	90 900	0.85



MCB Flange with ground Throat

Modified Flange with ground Throat

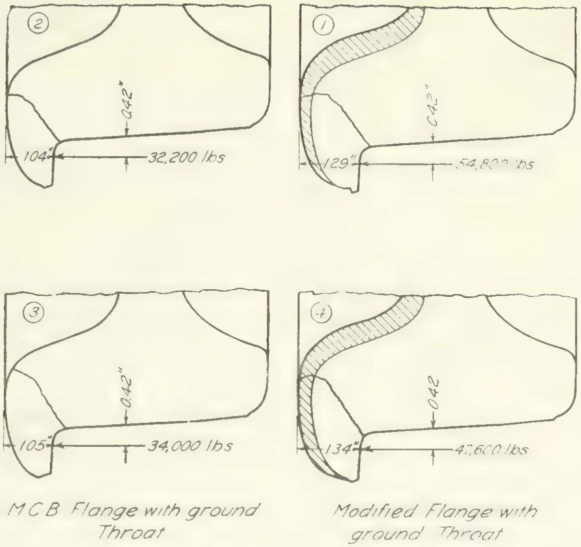
FIG. 37. 725-LB. M. C. B. WHEEL NO. 158 519. THROAT GROUND TO REPRESENT WORN FLANGE



MCB Flange as cast

Modified Flange as cast

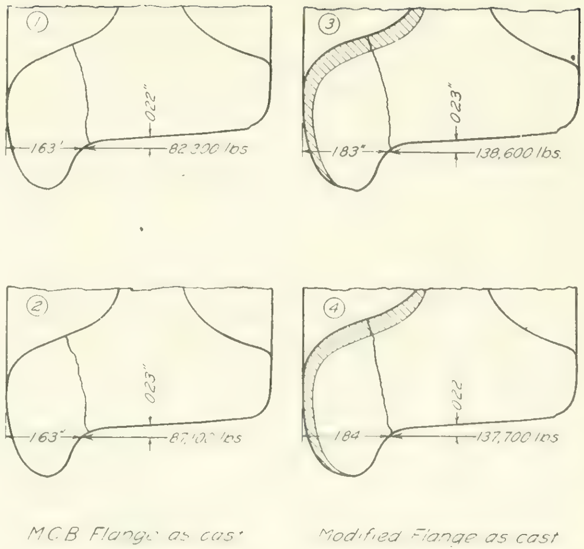
FIG. 38. 725-LB. M. C. B. WHEEL NO. 158 518. FLANGE AS CAST



M C B Flange with ground Throat

Modified Flange with ground Throat

Fig. 39. 725-lb. M. C. B. Wheel No. 158 521.
Throat Ground to Represent Worn Flange



M C B Flange as cast

Modified Flange as cast

Fig. 40. 725-lb. M. C. B. Wheel No. 158 520.
Flange as Cast

investigation should verify these results it would indicate that thickening the tread alone may not strengthen the flange, but on the contrary may decrease the lateral pressure required to cause failure. The explanation of this could probably be found in departure from the "ideal ratio" of flange to tread thickness.

18. *Effect of Thickening Flange and Tread Simultaneously.*—To investigate the effect of increasing the thickness of both flange and tread, two patterns were prepared in which half the circumference had the standard M. C. B. flange and tread and the other half the flange and tread thickened. In one case, represented by wheels Nos. 158 519 and 158 518 (Figs. 37, 38, and 46), the thickness of metal added to the flange was 3/16 inch at the extreme back of the flange, tapering down to nothing at the top of the flange; to the tread there was also added 3/16 inch of metal. In the other case, represented by wheels Nos. 158 521 and 158 520 (Figs. 39, 40, and 46), 3/16 inch of metal was added to the flange and 3/8 inch to the tread. Due to misplaced chillers during molding the actual additions of metal were slightly different from those stated. Two wheels were cast from each pattern, one of which was tested with the flange as cast and the other had the throat ground to represent a worn wheel before testing. Table 6 gives

TABLE 6

COMPARATIVE STRENGTH OF STANDARD M. C. B. FLANGE AND FLANGE OF WHEEL WITH THICKENED FLANGE AND TREAD

Wheel No.	Distance of Point of Application Below Tread in.	Ultimate Strength of Standard M. C. B. Flange lb.	Ultimate Strength of Flange of Wheel With Thickened Flange and Tread lb.	Ratio of Strength Thickened to Standard
158 519	0.41	29 200	31 300	1.07
(Worn Flange)	0.41	32 900	30 000	0.91
158 518	0.22	89 400	84 500	0.95
(Flange as cast)	0.22	87 100	91 400	1.05
158 521	0.42	32 200	54 800	1.70
(Worn Flange)	0.42	34 000	47 600	1.40
158 520	0.22	82 300	137 700	1.67
(Flange as cast)	0.23	87 100	138 600	1.59

the results of the tests (see also Figs. 37 to 40 inclusive), and in addition the ratio of the ultimate strength of the thickened flange and tread to the ultimate strength of the standard M. C. B. flange and tread.

In the wheels Nos. 158 519 and 158 518 in which 3/16 inch was added to both flange and tread, there was neither increase nor decrease in the ultimate strength due to the additional metal, as is shown by the fact that the ratio of strength of thickened to strength of standard flange was practically 1 both for the wheel having the worn flanges and for the wheel with the flange as cast. This seems to further verify the results obtained by thickening the flange and tread separately. It will be recalled that adding 3/16 inch to the flange alone increased the ultimate strength from 10 per cent to 96 per cent while adding 3/16 inch to the tread alone decreased the ultimate strength about 15 per cent. The figures here presented show that when 3/16 inch is added to tread and flange simultaneously no change occurs in the strength; and the conclusion might therefore be drawn that the increase in strength that would be expected due to the thickening of the flange was counteracted by a decrease of strength of approximately equal amount due to the addition of metal to the tread.

The results of the tests made on the other pair of wheels, Nos. 158 521 and 158 520, in which 3/16 inch was added to the flange and 3/8 inch to the tread, show that an appreciable increase in the ultimate strength of the flange was caused by the addition of metal in this manner. The ratios given in the table show a minimum increase in flange strength in this case of 40 per cent while the maximum was 70 per cent. These increases in strength were obtained by the addition of metal equivalent to approximately 37 lb., or 5.1 per cent when applied to a 725-lb. wheel.

These results indicate that the maximum flange strength of the chilled iron wheel has not yet been reached, but on the contrary that the strength of the flange can be materially increased by the proper placing of additional metal in flange and tread. They further show that the ratio of the strength of thickened to strength of standard flange is practically the same for new and worn flanges in each pair of wheels tested, indicating that when metal is added simultaneously to both flange and tread, as was done in these wheels, the advantage of the additional metal does not disappear as the wheel approaches the condemning limit through flange wear.

VII. Summary of Conclusions

19. *Combined Effect of Mounting, Static Load, and Flange Pressure.*—The results of the tests made on the 33-in. 840-lb. Arch Plate wheel to determine the combined effect of mounting, static load, and flange pressure lead to the following conclusions:

> (1) In mounting the wheel on the axle the largest tensile strains in the wheel occur on the inner face in a tangential direction at the bore and their magnitude decreases as the radial distance from the bore increases. The radial strains on the inner face reach a maximum in compression at the bore; as the radial distance from the bore increases the compression decreases to zero, the strains becoming tensile at about the mean radius of the core; the tension reaches a maximum value just before the intersection of the inner and outer plates; beyond this point the tension decreases to zero and the strain again becomes compressive. This distribution of the strains plainly suggests bending due to the curvature of the section in the region of the core.
>
> (2) On the outer face of the wheel the strains in a tangential direction due to mounting are a maximum in tension at the bore; as the radial distance from the bore increases, the tension decreases up to a point near the intersection of the inner and outer plates, beyond which a slight increase in tension occurs. In the radial direction, over the region investigated, the compression found near the bore becomes larger as the radial distance from the bore increases, and reaches its maximum value slightly beyond the mean radius of the core; after passing this point it rapidly decreases to practically zero. This strain distribution in the radial direction plainly indicates a bending action combined with the direct thrust due to mounting.
>
> (3) The largest tensile strains in the wheel caused by mounting are found in the hub metal near the axle. Although apparently of large magnitude when expressed in terms of "corresponding simple stresses," these are steady strains, that is, the metal is not subject to fluctuating or repeated internal stresses

that would produce alternate strains and removal or reversal of strain in the material. The character of the strains, together with the backing provided the metal most highly strained, probably permits the occurrence without injury to the material of greater strains than would normally be possible in metal subjected to simple tension or compression.

(4) The application of the normal static load of service to the wheel does not materially alter the strains already present on the outer face of the wheel through mounting. On the inner face the bending of the curved plate in the core region becomes more pronounced, as indicated by tensile strains in a radial direction. These tensile strains are smaller than those due to mounting found in the tangential direction near the axle. If this type of wheel were called upon to carry excessively heavy static loads, the tensile strains might become significant, but for normal service those occurring are not in themselves serious, and they are modified and reduced by the application of flange pressure.

(5) The combined effect of mounting, static load, and flange pressure, approximating the maximum occurring in service, in general results in tensile strains in both radial and tangential directions on the outer face. The maximum tension occurs near the junction of the inner and outer plates, and is in a radial direction. The tensile strains produced throughout the outer face by flange pressure, although subject to some repetition, may safely be considered as steady strains, because the number of repetitions of strain that would occur in the normal life of a wheel — due to the occurrence of sufficiently high flange pressure caused by traversing curves of small enough radius at high enough speed — would be relatively insignificant compared with the millions of repetitions found necessary to cause failure under similar conditions in investigations of repeated stress. In themselves, the strains caused by the combination of mounting, maximum static load, and maximum flange pressure are not of serious magnitude; but if to these be added strains caused by severe and excessively-prolonged brake application, the combined strain might then become large enough to cause incipient cracks leading to failure. On the inner face of the wheel over the region investigated, the strains in the tangential direction due to the combined effect of

mounting, maximum static load, and flange pressure differ but little from those due to mounting alone. In the radial direction the strains are all compressive. The strains on the inner face in both radial and tangential directions are of relatively little importance.

20. *Ultimate Strength of Wheel Flanges.*—The conclusions to be drawn from the tests made to determine the ultimate strength of the wheel flange may, within the limits of these tests, be summarized as follows:

(1) When the flange pressure is applied to wheels with badly worn flanges on a radius midway between the points where two adjacent brackets enter the tread, the ultimate strength of the flange is an inverse function of the distance of the point of pressure application from the tread, that is, the greater the distance the smaller the ultimate strength.

(2) This inverse relation does not hold in the case of new or slightly worn flanges, neither does it hold in the case of either new or worn flanges if the point of pressure application is on the same radius as that on which a bracket enters the tread. In this case it was found in some of the tests that the bracket apparently strengthened the flange, while in others a weakening of the flange was observed, these conditions indicating a complex supporting effect due to the bracket.

(3) The addition of 3/16 inches of metal to the back of the present standard flange, as proposed by the Association of Manufacturers of Chilled Car Wheels, increases the ability of the flange to withstand side thrusts. This increased strength was plainly shown in the tests of five out of six of the specially prepared wheels, in which one-half of the circumference had the present standard M. C. B. flange and the other half the proposed additional flange thickness. In the case of the one wheel apparently showing decreased flange strength as a result of the additional metal, the flange and tread proportions were irregular due to a misplaced chiller during molding, and this fact makes it inadvisable to use the results from this wheel. The other five wheels showed an increased flange strength due to the additional

metal of from 10 to 90 per cent in the case of badly worn flanges, and from 26 to 49 per cent in the case of slightly worn or new flanges.

(4) In the case of the wheel having one-half of the circumference cast with the standard M. C. B. tread and the other half with the 3/16 inches of additional metal on the inner and under side of the tread, the results were negative, inasmuch as the half with the additional metal showed decreased ability to withstand side thrust. Although tests on one wheel cannot be taken as conclusive, if further tests confirm these results, the indication would be that it is possible to add metal in such a way that not only no increased flange strength results but a decreased flange strength may actually be caused. If this be taken as proved, then there probably exists a relation or ratio between flange and tread thickness which will result in a maximum flange strength for a particular thickness of either flange or tread. In that case, if a wheel be cast with the ideal ratio of tread to flange thickness for, say, a given thickness of flange, the addition of metal to or the subtraction of metal from the tread will not only not increase the flange strength but may even decrease it.

(5) The simultaneous addition of 3/16 inches of metal to both flange and tread was not found to alter the ability of the flange to withstand side thrust. As the addition of 3/16 inches to the flange alone gave added strength, this result would tend to confirm the conclusion reached in (4). The simultaneous addition of 3/16 inches of metal to the flange and 3/8 inches to the tread was found to result in an appreciable increase in the ultimate strength of the flange. The weight of metal added was about 37 pounds or 5.1 per cent of a 725-lb. wheel, whereas the increase in flange strength resulting therefrom ranged from 40 to 70 per cent. An important feature of this case is that the advantage of the additional metal apparently does not disappear as the wheel approaches the condemning limit on account of flange wear.

(6) The general conclusion to be drawn from the results of these tests is that the flange strength of the chilled iron wheel can be increased to keep pace with probable future service requirements by a proper placing of additional metal in the tread and flange.

APPENDIX A

Views of Chilled Iron Wheels After Application of Flange Pressure

Figs. 41 to 46 inclusive are reproduced from photographs of the inner faces of the several wheels after they had been tested for ultimate flange strength. These views are presented mainly to show the location at which the loads were applied with respect to the brackets; since the maximum diameter of the wheel is 35 inches, they also show the approximate size of the pieces broken out. The numerals painted on the wheels in the neighborhood of the fractures are identification numbers, and serve to connect the illustrations with the data and figures given in the body of the text.

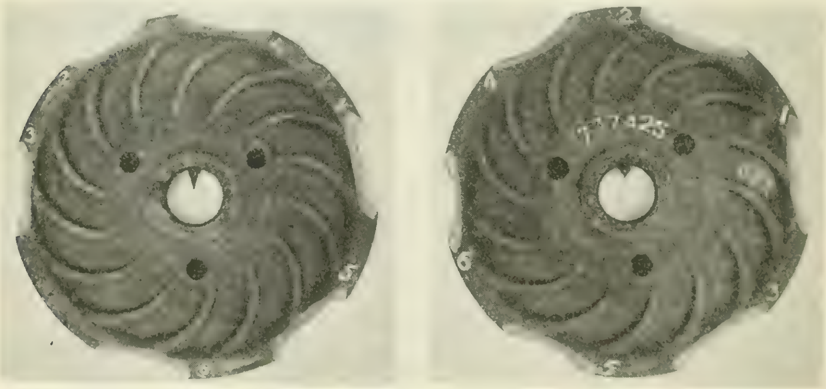


Fig. 41. Inner Faces of Wheels Nos. 737 401 and 737 425 After Fracture of Flange

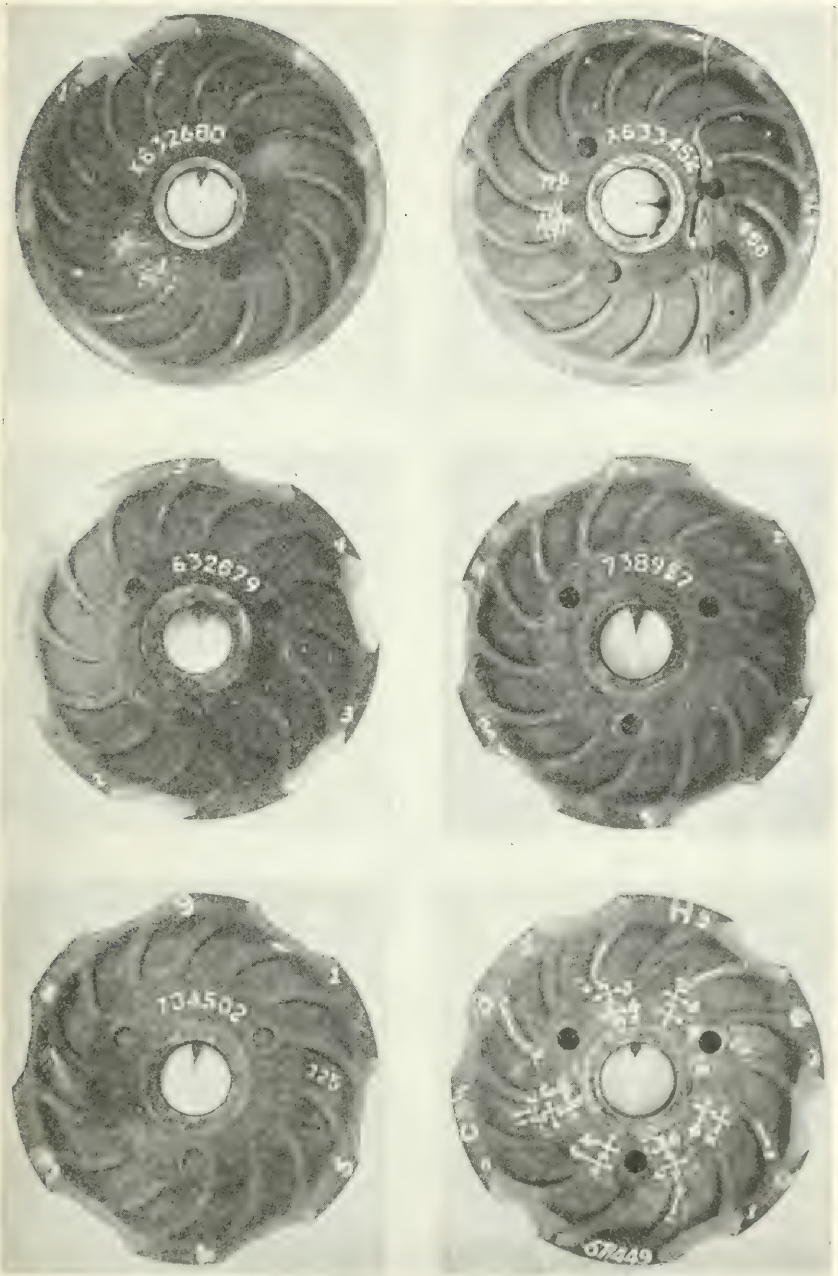


FIG. 42. INNER FACES OF WHEELS NOS. 632 680, 633 452, 632 679, 738 957, 734 502, AND 671 449 AFTER FRACTURE OF FLANGE

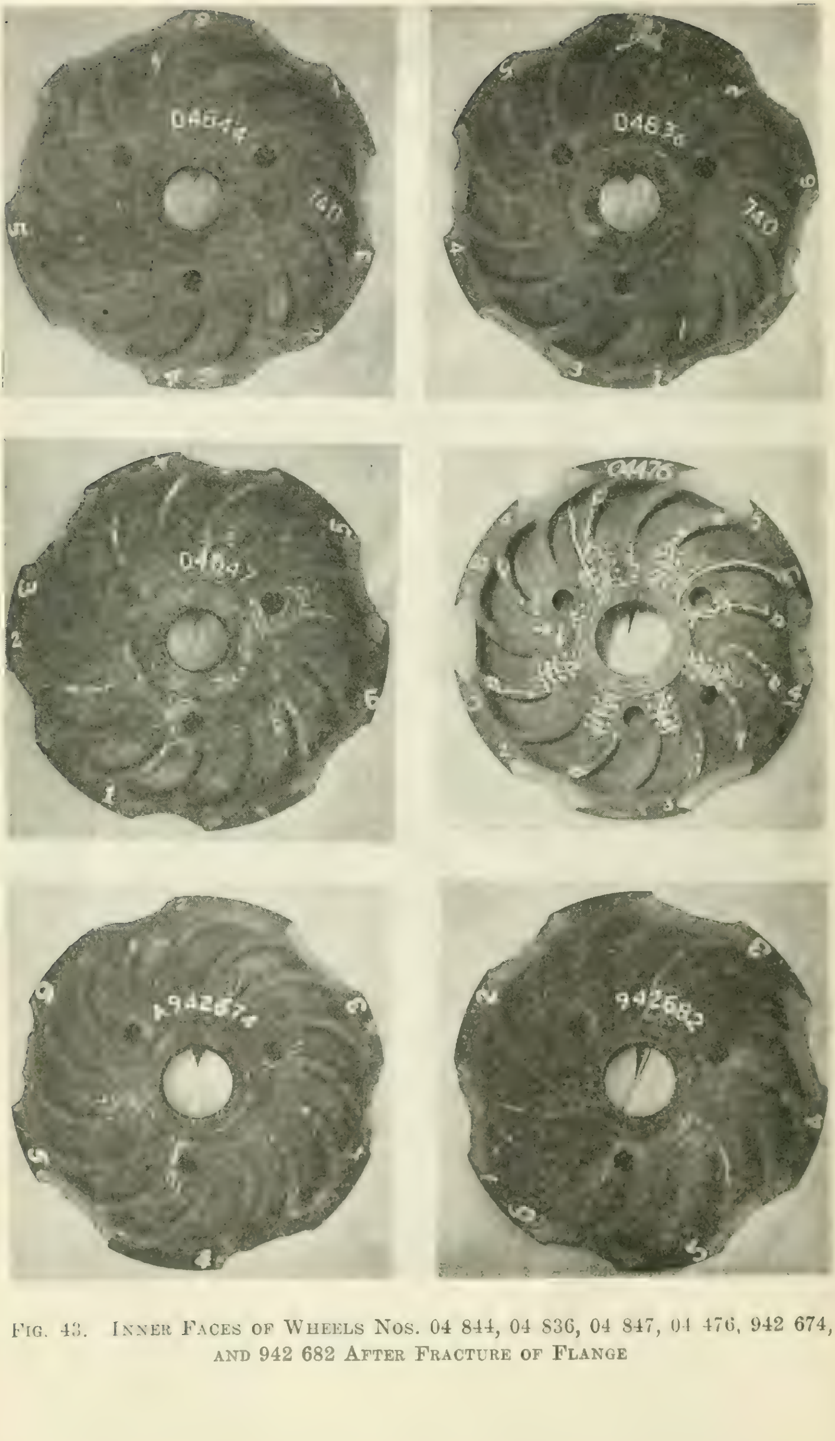


Fig. 43. Inner Faces of Wheels Nos. 04 844, 04 836, 04 847, 04 476, 942 674, and 942 682 After Fracture of Flange

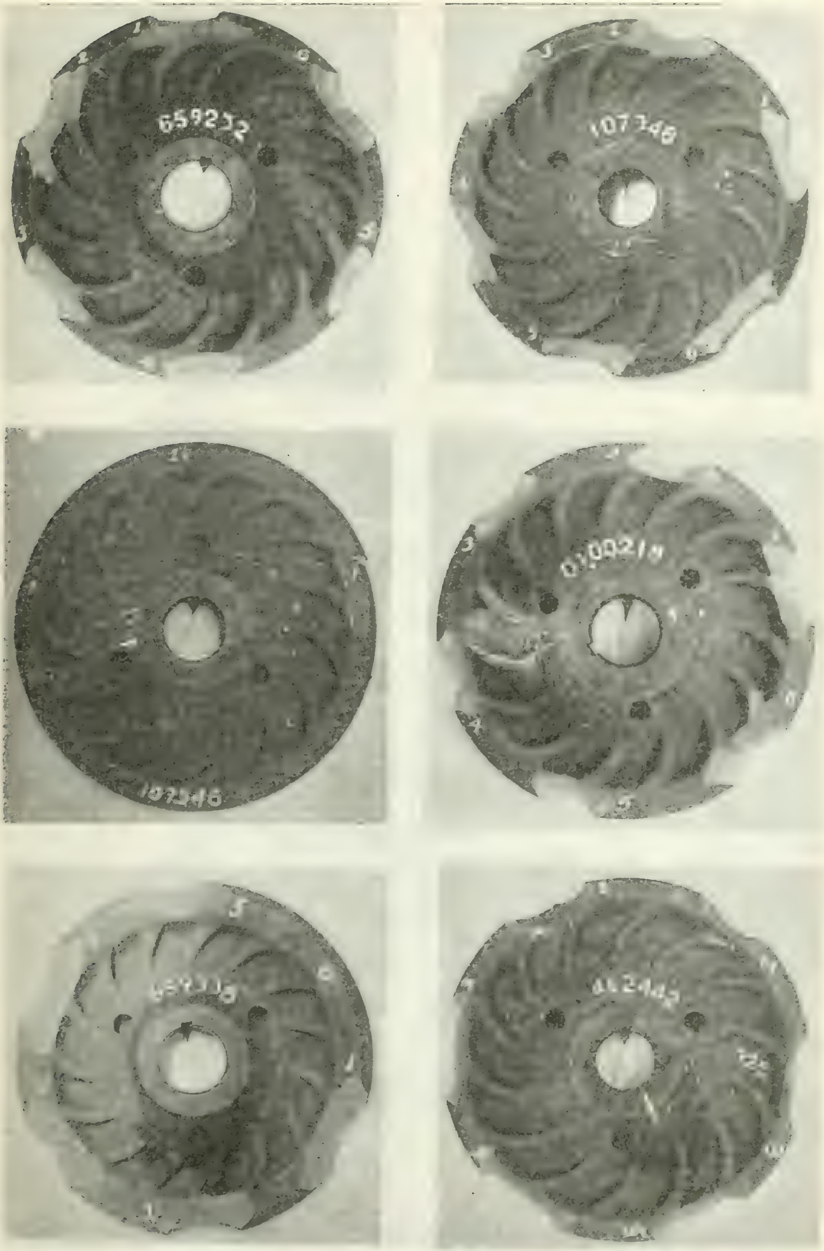


Fig. 14. Inner Faces of Wheels Nos. 659 232, 107 348, 107 346, 103 219, 659 338, and 462 442 After Fracture of Flange

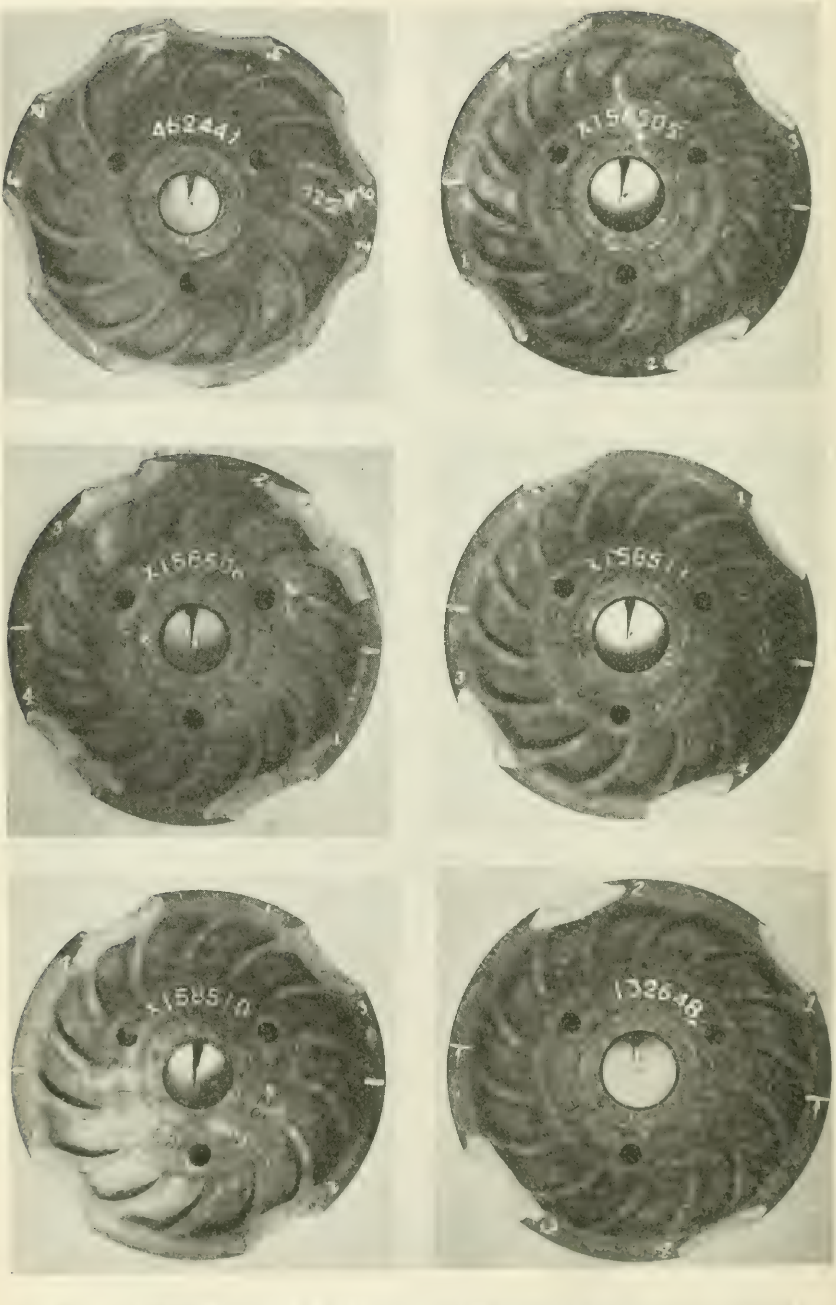


Fig. 45. Inner Faces of Wheels Nos. 462 441, 158 505, 158 506, 158 511, 158 510, and 132 648 After Fracture of Flange

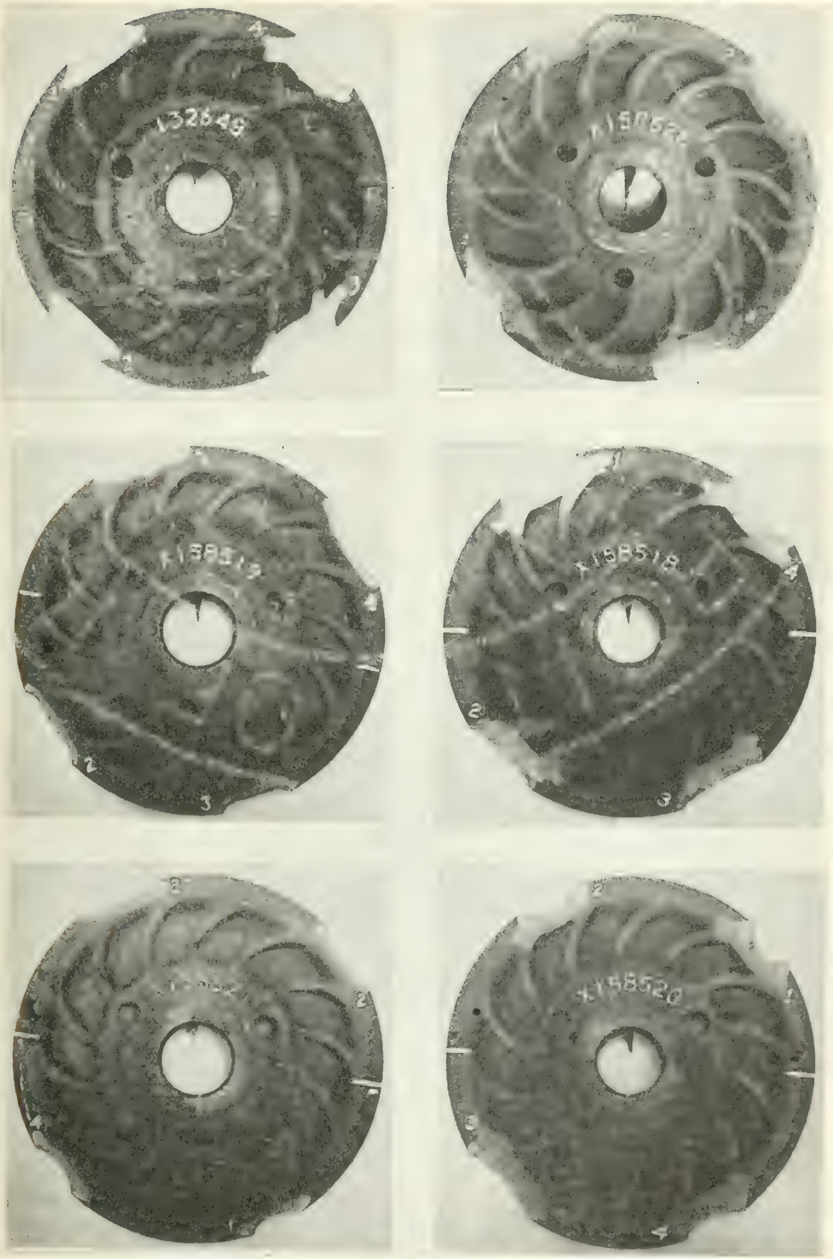


Fig. 46. Inner Faces of Wheels Nos. 132 649, 158 525, 158 519, 158 518, 158 521, and 158 520 After Fracture of Flange

LIST OF
PUBLICATIONS OF THE ENGINEERING EXPERIMENT STATION

Bulletin No. 1. Tests of Reinforced Concrete Beams, by Arthur N. Talbot. 1904. *None available.*

Circular No. 1. High-Speed Tool Steels, by L. P. Breckenridge. 1905. *None available.*

Bulletin No. 2. Tests of High-Speed Tool Steels on Cast Iron, by L. P. Breckenridge and Henry B. Dirks. 1905. *None available.*

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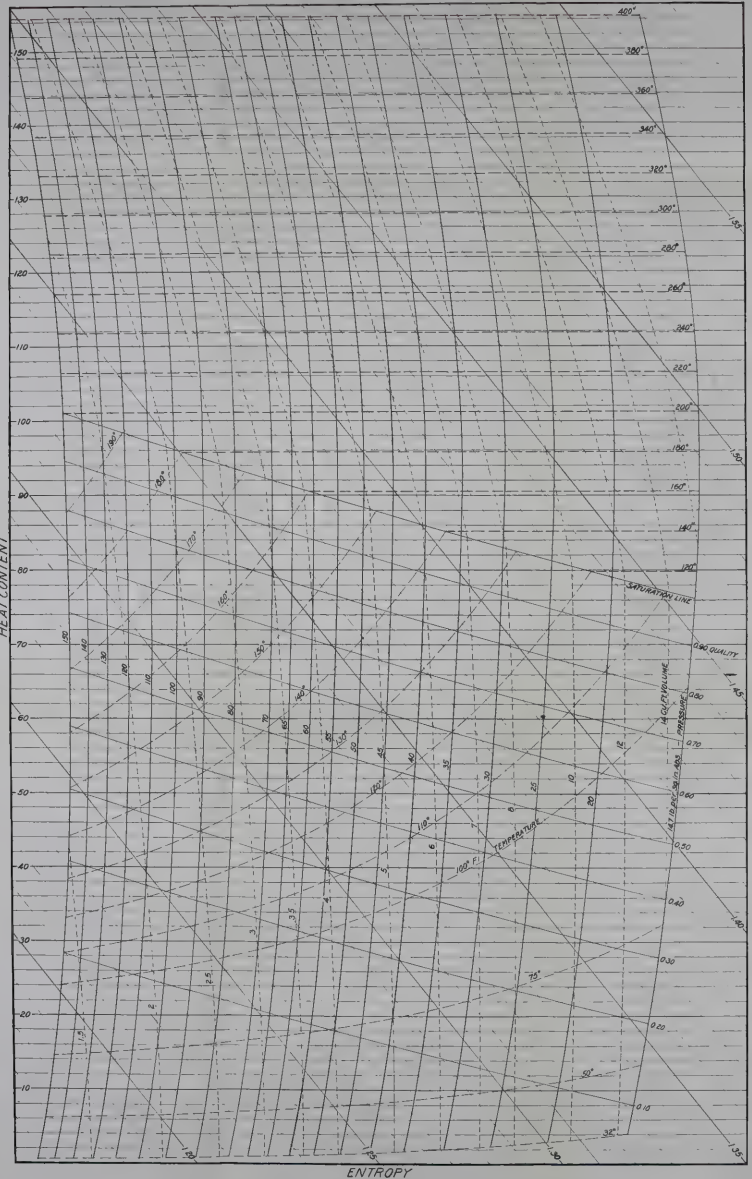

Chart 1. Entropy—Heat-Content Diagram for Air-Steam Mixture; 5 per cent Steam

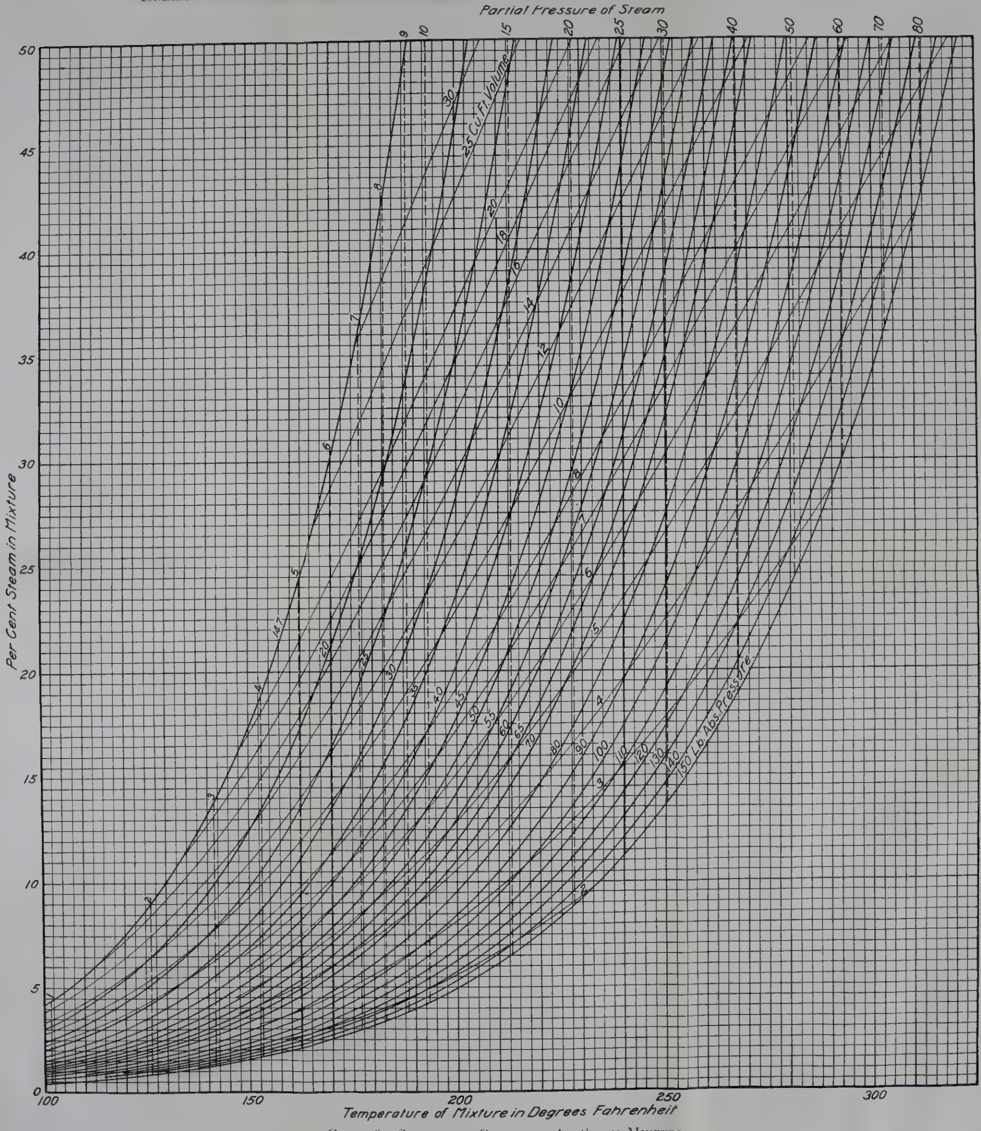

Chart 9. Saturation Chart for Air-Steam Mixtures

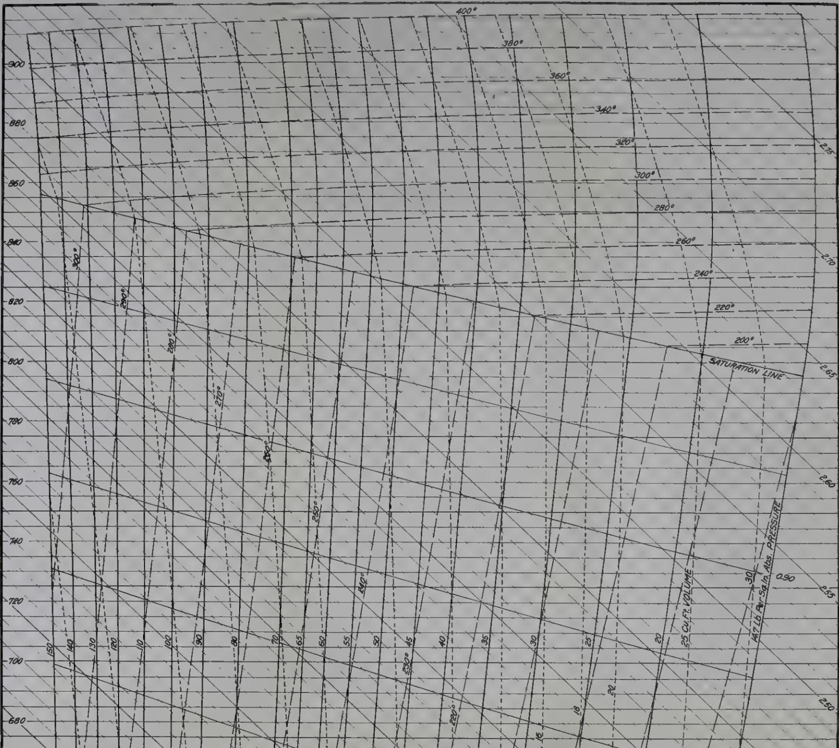

Chart 8. Entropy-Heat-Content Diagram for Air-Steam Mixture; 40 per cent Steam

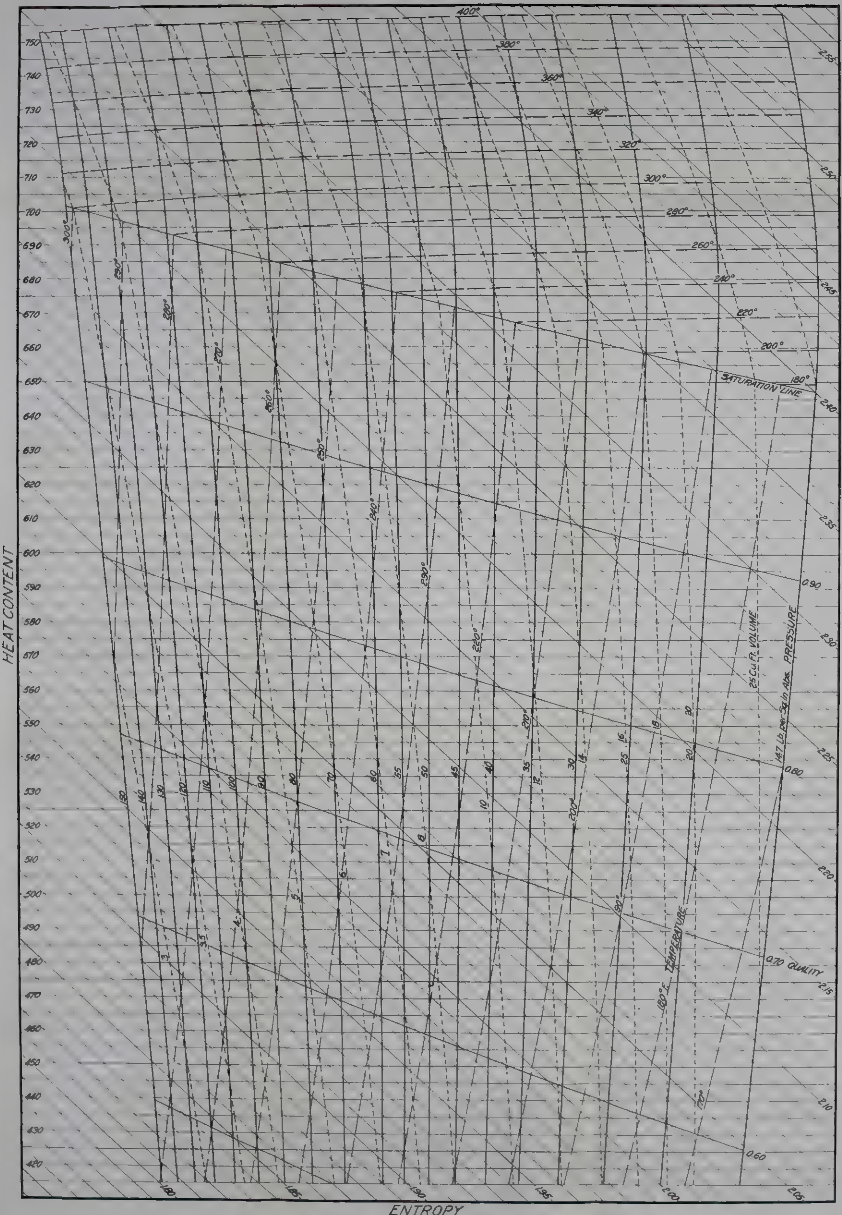

Chart 7. Entropy–Heat-Content Diagram for Air-Steam Mixture, 35 per cent Steam

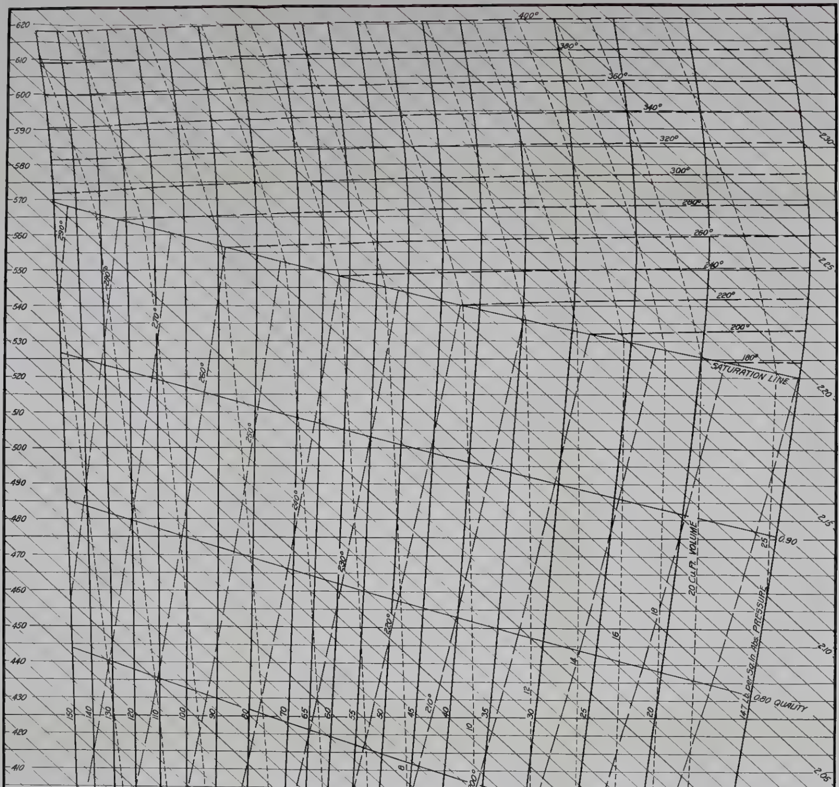

Chart 6. Entropy Heat-Content Diagram for Air-Steam Mixture; 30 per cent Steam

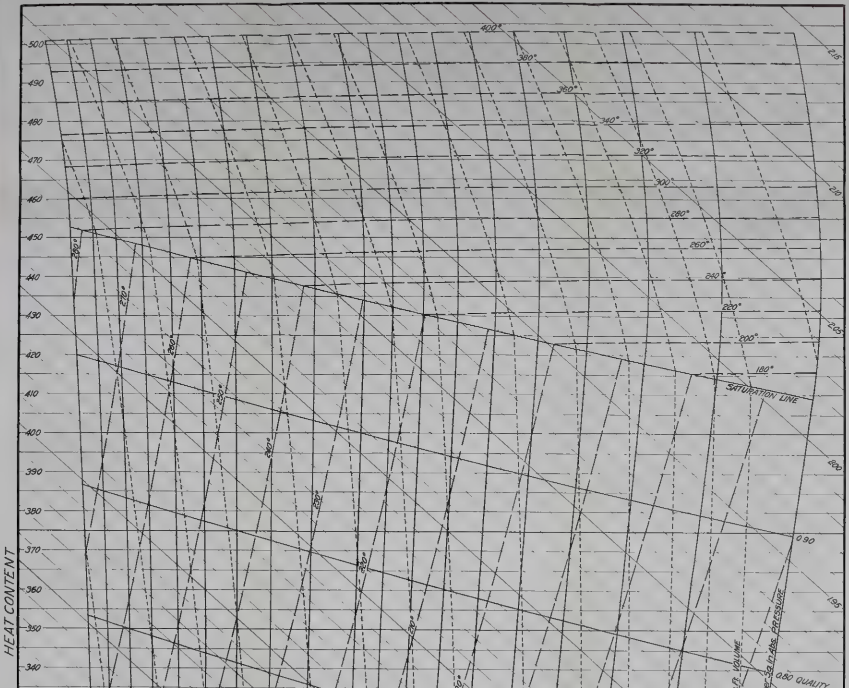

Chart 5. Entropy Heat-Content Diagram for Air-Steam Mixture, 25 per cent Steam

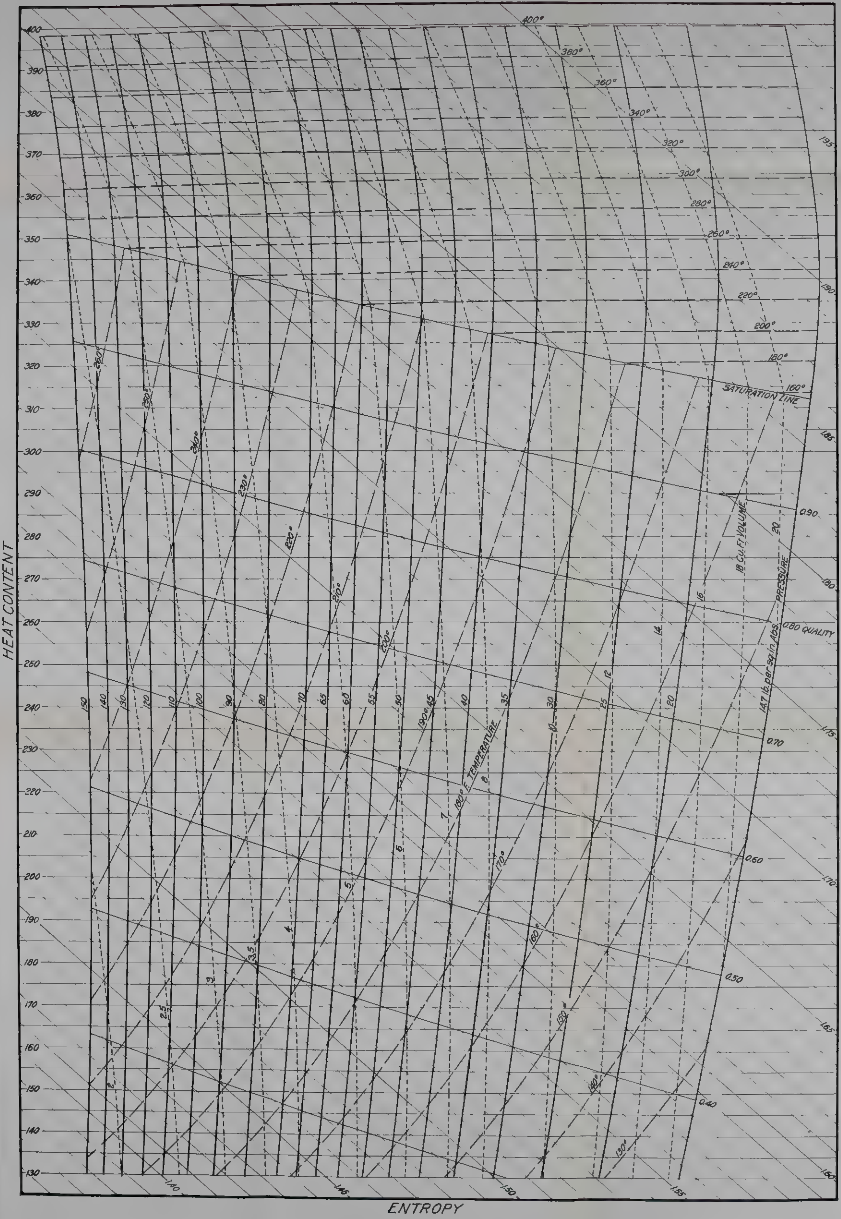

Chart 4. Entropy—Heat-Content Diagram for Air-Steam Mixture; 20 per cent Steam

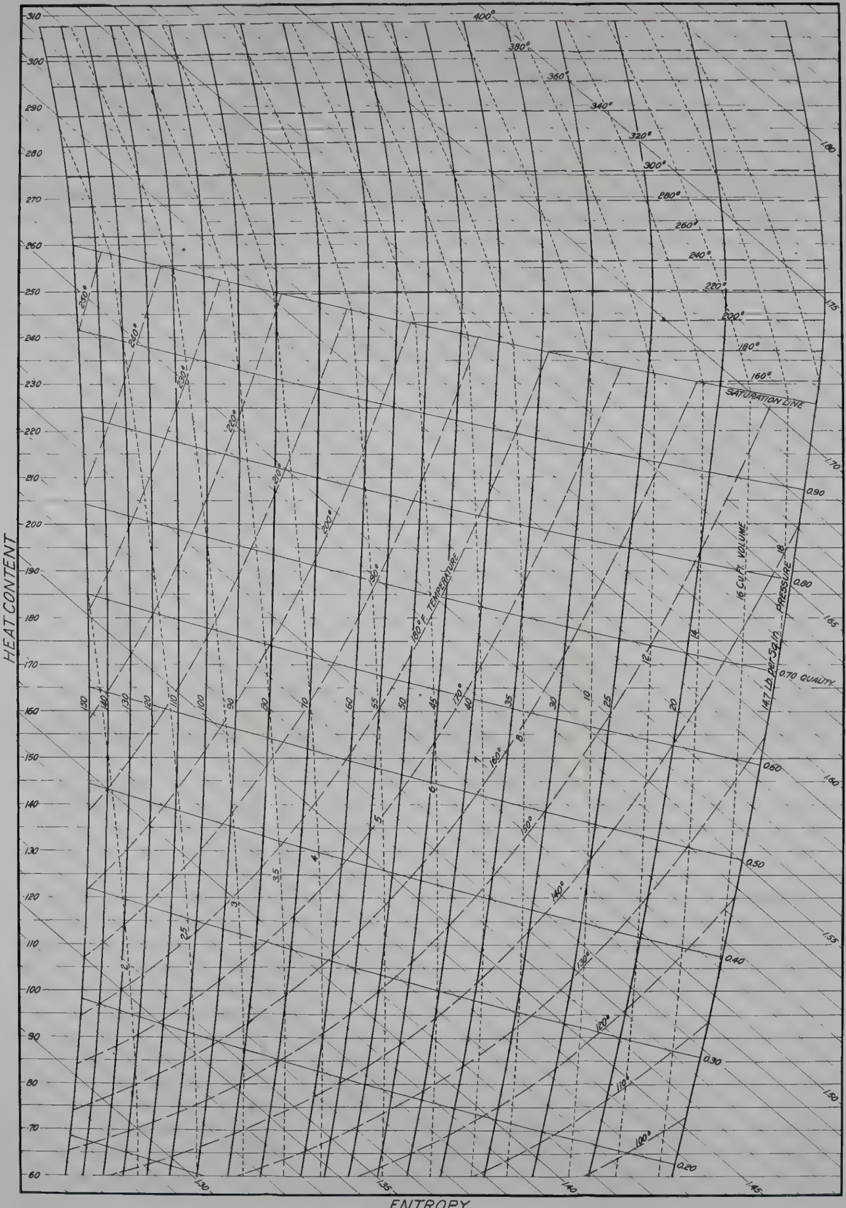

Chart 3. Entropy—Heat-Content Diagram for Air-Steam Mixture; 15 per cent Steam

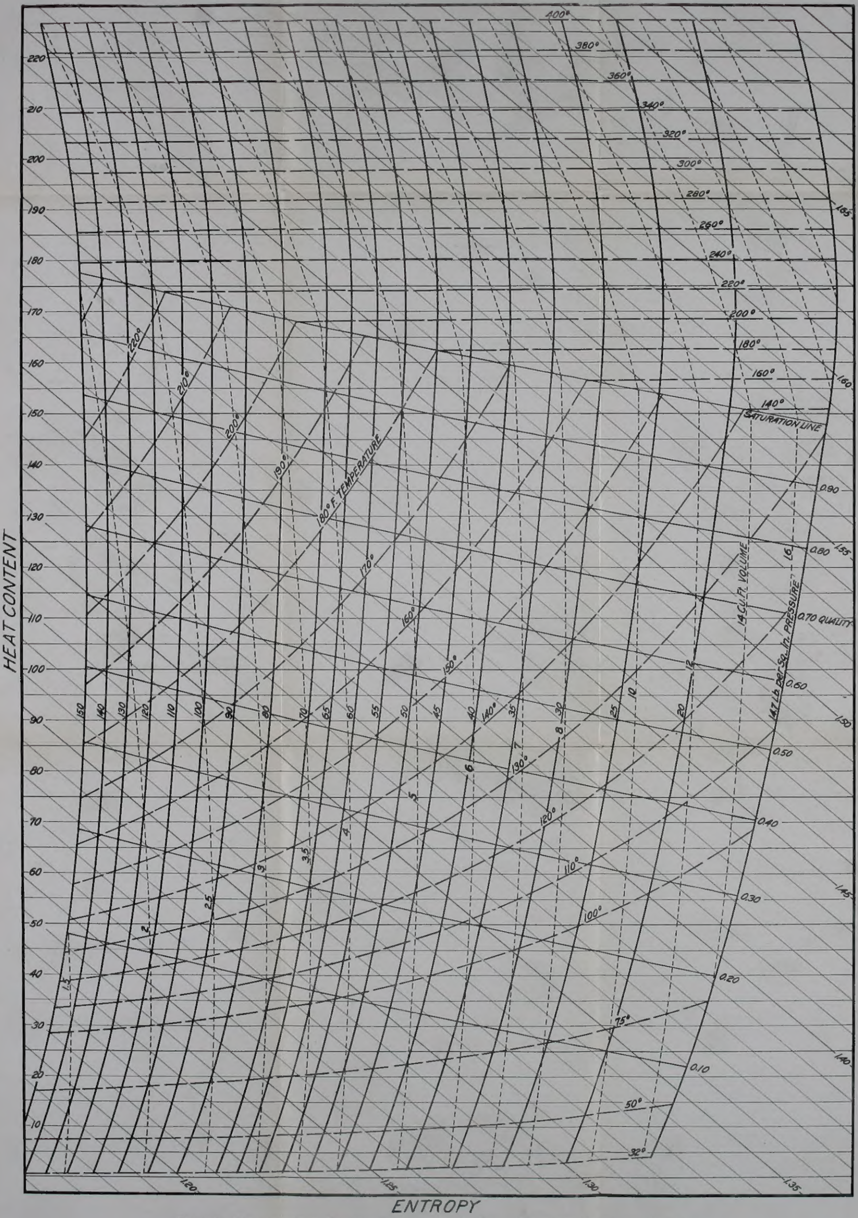

Chart 2. Entropy—Heat-Content Diagram for Air-Steam Mixture; 10 per cent Steam

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